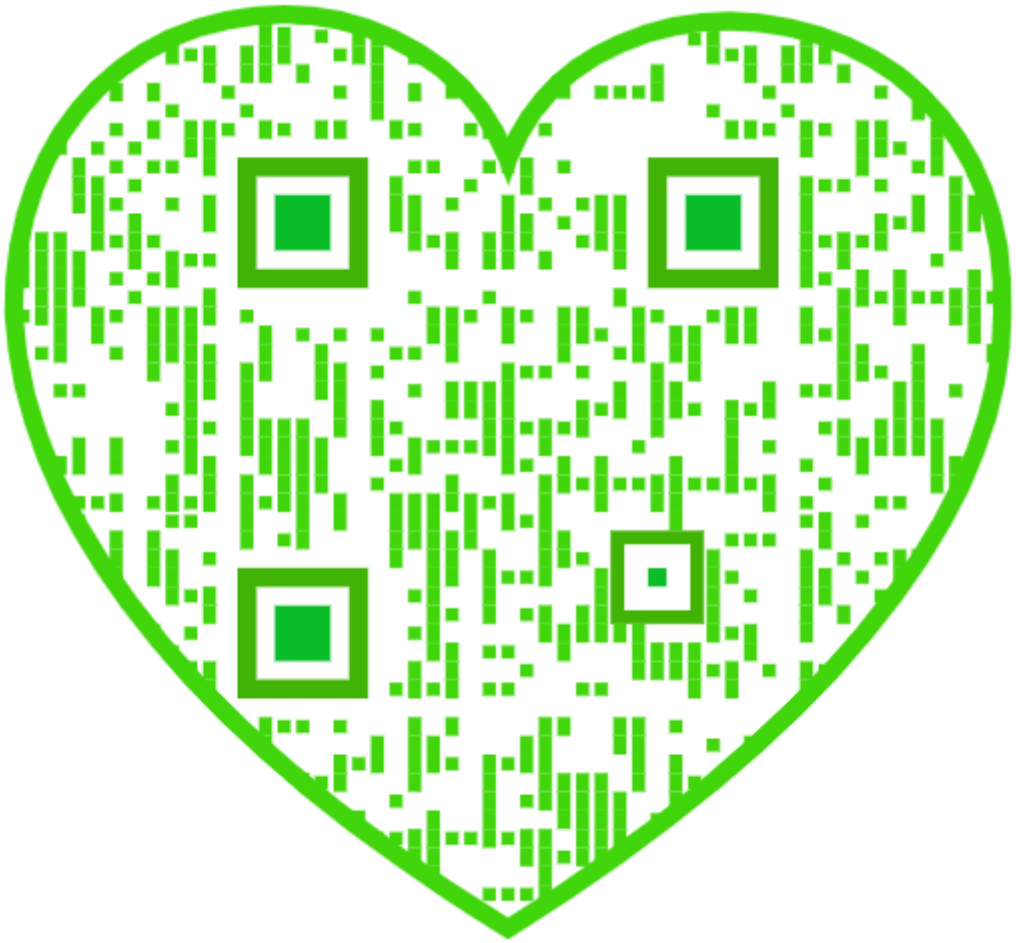

Eugene T. Skinkle

F MAKING

PRACTICAL ICE MAKING

AND

REFRIGERATING

A PLAIN, COMMON SENSE SERIES OF PAPERS ON THE CONSTRUCTION AND OPERATION OF ICE MAKING AND REFRIGERATING PLANTS AND MACHINERY

BY

EUGENE T. SKINKLE

"THE BOY"

CHICAGO
H. S. RICH & CO.
1897

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PRACTICAL ICE MAKING

AND

REFRIGERATING.

INTRODUCTORY.

TWENTY YEARS OF REFRIGERATION.

TWENTY years ago this November (1896) a bearded, pleasant faced man with a huge roll of drawings under his arm entered the offices of Crane Bros. Manufacturing Co. (now Crane Co.), of Chicago, and in a quiet, gentlemanly manner asked a boy at the first desk "if ice machines could be manufactured in that shop." The boy clutched the desk to prevent falling off his high stool, and after taking a long look at the stranger to assure himself that the man was mentally sound, and that it was safe to venture outside the railing, took the stranger to the superintendent of the machinery department and witnessed another shock when the question was repeated to the superintendent. The stranger was the late David Boyle. "The Boy" will endeavor to interest the reader with a partial review of the intervening twenty years, the greater portion of which time he has devoted to ice making

and refrigerating; and as an honest confession is said to be good for the soul, the writer will frankly confess at the start that he has lots to learn yet in the fields in which he is trespassing.

Of the late David Boyle it may be truthfully said that he was the father of successful artificial refrigeration and ice making; for although many had spent years of toil and vast sums of money in experiments in the art before he entered the field, yet to David Boyle is due the honor of having accomplished more in the advancement of the science than the combined results of the efforts of all of his predecessors.

To the skill and energy of many other "old timers" I would offer deserved tribute, among whom I would name Mr. Beeth, Thomas L. Rankin, D. L. Holden, Mr. Kilgore, Mr. de Coppet and Oscar Burnham, men who have spent their time and their dollars in the cause and without adequate returns from the public, who have benefited by their brains and ingenuity.

My own first knowledge of an ice machine dates back to the winter of 1875-76, when a little "wheel-barrow" machine was constructed and erected in Crane's shop. It was a 2-ton plate plant, with horizontal, slide-valve engine and vertical, single-acting pumps, plate freezing tanks, submerged condenser—as I now consider it, a pretty generally crude

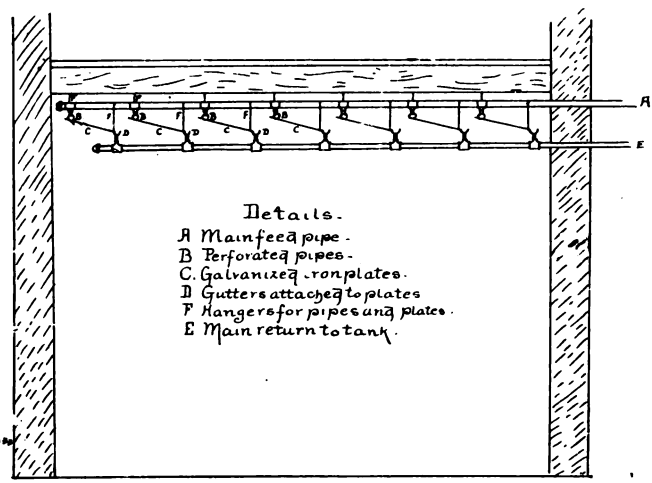
construction. The plant was provided with an apparatus for distilling anhydrous ammonia from the aqua (and a number of the first machines built at Crane's were so provided). Two batches of ice were made with the machine before it was shipped, and I have never seen better or clearer ice made since. In general design that little plant did not differ essentially, with the possible bare exception of some slight modifications in adaptation of the freezing plates, from plate plants of to-day; and in principle, proportions of surfaces and general design of parts, the little plant would compare favorably with the average present practice. In detail and construction present practice would far outclass the crude little plant.

For about two years D. Boyle & Co. built ice machines only, plate plants like the original. In 1877 (I believe) their first refrigerating machine was constructed, under contract with Mr. Harry Bemis, for the Bemis & McAvoy Brewing Co., of Chicago. This was rated as a 20-ton machine, and was used for cooling wort on a Baudelot cooler. The plant consisted of a pair of 10×18-inch vertical, single-acting pumps, operated by a horizontal Brown engine; a sweet water tank about 14×12×8 feet, containing a double system of 1-inch coils, one set above the other, with separate outside manifolds. An agitator was placed in the tank, with paddles between the coils to pre-

vent the sweet water from freezing solid in the tank. The sweet water was cooled and pumped through the Baudelot and returned to the tank. The condenser was submerged in a tank ten feet square and six feet deep. There were eighteen coils, twelve pipes high, seven and a half feet long, one inch extra-strong pipe. The connections were not such as your critical engineer of to-day would approve, but they answered the purpose eighteen years ago. The plant cooled beer in the Bemis & McAvoy brewery for several years, and after the purchase of a 50-ton Boyle machine for the brewery, was sold to N. K. Fairbank & Co.'s soap works in Chicago, where it has done good service and stands to-day ready to be used in case of emergency. Harry Bemis took as much interest in this original Boyle refrigerating machine as he has ever done in enterprises taken up by him. He put on overalls and worked with the gang and had a table set in the brewery for all of the workmen, and no guest at the Richelieu enjoyed better fare than Boniface Bemis' hospitality served in the brewery in 1877—with frothy beer on the side.

About one year later contracts were taken for the complete refrigeration of Heim's brewery in East St. Louis, Ill., and Frank Fehr's brewery in Louisville, Ky., and the partial refrigeration of the Best (now Pabst) brewery in Milwaukee, Wis. The Fehr and Best con-

tracts were started earliest, and in both cases 25-ton machines were furnished, 14×30-inch engines and 10×20-inch single-acting pumps, all vertical; brine tanks 16×13×8½ feet, with thirty-eight coils, twelve pipes high and eleven feet long; submerged condensers 10×10×6½, with twenty-two coils, twelve pipes high and seven and one-half feet long. Instead of piping



the cellars for brine circulation with sufficient pipe to provide the requisite cooling surfaces, perforated pipes were run at intervals of about three feet apart, and sheets of galvanized iron were suspended below the pipes to catch the cold brine from the perforations and spread it over the surface of the plates and run it into gutters at the opposite end of the plates, from which gutters the brine was returned to the

brine tank. This system was later abandoned, owing to the perforations in the feed pipes becoming clogged and preventing an even distribution of the cold brine over the plates; and the cellars were piped with sufficient piping to furnish the necessary cooling surfaces with confined brine circulation (about one lineal foot of 1-inch pipe to ten cubic feet of space cooled). The Heim's plant in East St. Louis was not fitted up with the exposed brine circulation, but was originally piped for confined brine circulation, and remains as it was originally designed. This exposed, or plate, circulation was the origin of the open gutter circulation used in some isolated plants at present.

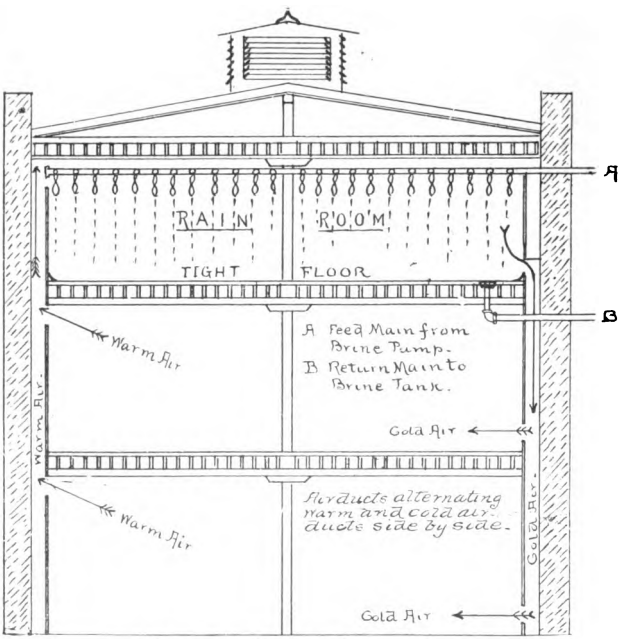
Benefiting by the experience gained at Fehr's and Best's, the confined brine circulating system was adopted and installed in upwards of 200 breweries, packing and cold storage houses in a period covered by ten years' time.

One of the novelties in the way of refrigeration in the early '80's was a plant erected for Henry Ames, St. Louis, Mo. Mr. Ames' business was curing hams, and a small plant—about ten tons refrigerating capacity—was specially adapted to an original idea of his own. Mr. Ames conceived the idea of pickling hams by means of a continuous circulation of cold pickle through the tiers of hams in the pickle vats. The small machine was

used to keep the pickle at a uniform temperature, and a small brine pump was used to keep a continuous circulation of the pickle through the vats. It is a matter of record that the process was a complete success, Mr. Ames having testified "that he never had a spoiled or sour ham while using this process." The writer regrets that he is unable to give the recipe for the pickle, but the system can be readily adapted to the pickle vats in any curing room; and it is patent that circulation will facilitate the process of pickling meats. Mr. Ames retired some years since, said to be "well fixed" financially.

Another peculiar and effective system of refrigeration, applied years ago, was the indirect system adopted by N. K. Fairbank & Co., of St. Louis, Mo. In this plant a regulation brine system as regards machine, brine tank and brine pump was used, but differing from any other plant the writer knows of in the following respects, viz.: A "rain room" was located at the highest point of the plant, above the rooms to be cooled. This room was air-tight on all sides and above, and was provided with a water-tight floor. Air ducts led from a little above the floor line down to the rooms below, and warm air ducts from the rooms below up to within a short space of the ceiling of the "rain room." The "rain room" was piped with perforated pipes, from which innumerable bunches of rope yarn were hung.

The cold brine was pumped from the brine tank to the perforated pipes in the "rain room," and the result was a continuous artificial cold rain storm, cooling the air in the room, which in turn fell by reason of gravity



Indirect Brine System

to the room below, forcing the warmer current of air upward and causing a continuous rapid circulation of air through the ducts, to and from the rooms below. The "rain" was caught by the water-tight floor and conveyed through suitable pipes back to the brine tank.

This system offers the inducement of a fresh sea breeze at home, and to the thinking mind affords a substitute for our wives' favorite summer resort. Its drawback in certain lines of practical cold storage is the humidity necessarily imparted to the air in circulation, and that other objection, existing in all systems of open brine circulation, ultimately stinking brine from the noxious gases absorbed by the brine from the air. It cools and purifies the air so long as the brine remains reasonably sweet; but when the brine becomes surcharged with impurities from the air, it will cease to be a desirable matter to bring the bad smelling brine in contact with the purer air.

CHAPTER I.

COOLING SURFACES AND CIRCULATION.

MANY articles have appeared in numerous trade journals, from time to time, treating upon the subject of practical refrigerating surfaces for various cold storage purposes. The columns of *Ice and Refrigeration* teem with practical suggestions, comprehensive cuts and valuable hints for information of the progressive engineer. That the inventor has given his attention to the subject is evidenced by numerous files in the patent office of the nation. A writer on this subject may, therefore, expect to be accused of borrowing the ideas of others, should he attempt to review the progress made in this branch of the art in the light of personal observations. However much has been said on the subject, no attempt has been made, in the knowledge of the writer, to concentrate and review the various systems which have been and are now in vogue. It cannot be expected that any one person can keep track of and be in a position to fully cover the progress made in such a vast field unless he devotes his entire time to the research and invites unlimited corre-

spondence on the subject. The writer will, therefore, only touch upon such systems as have come within his own observation and with which he is familiar, at the same time requesting the interested readers to fill in any uncovered space in the record by sending to the editor of *Ice and Refrigeration* the record of any novel system which may have come within the experience or observation of themselves. Thus will we all be benefited by a mutual interchange of experiences, through the medium of the columns of a live and progressive journal, read all over the globe, and capable of collecting invaluable information for its readers.

Cooling surface systems may be divided into two classes: The direct system, where the surfaces are placed within the chambers to be refrigerated; and the indirect system, where the surfaces are located outside of the chambers and the cold air conveyed into the chambers by means of natural gravity or by mechanical force (fans, blowers or air pumps). The indirect system was the original system of cold storage refrigeration in use in this country, illustrated by the old and rapidly disappearing system of placing natural ice chambers above the cold storage chambers, connecting the two chambers together with cold and warm air ducts, and causing a circulation of the air by means of a difference in gravity. The writer is aware that in some

isolated instances ice has been placed directly in the cold storage chambers, but such practice is so primitive as to be unworthy of even a passing consideration.

One of the first examples of indirect refrigeration, by mechanical force, was fully illustrated in an early catalogue of the Sturtevant Blower Co. The apparatus consisted of a Sturtevant blower, a tank or pan containing a system of air pipes in flat coil form, and branch pipes from the coils to the cooling rooms or chambers. The coils in the pan were surrounded with ice and salt, and the air from the blower passed through the coils and branch pipes to the cooling chambers, being cooled by exposure to the chilled surfaces of the coils. The system was an improvement over the ice chamber system, in that it condensed the moisture from the air in the cooling coils and delivered dry air to the cooling chambers. These two systems may be said to aptly illustrate the principles of the gravity and mechanical-force indirect systems utilized in connection with mechanical refrigeration at the present time, substituting merely ammonia or brine for the ice element of the systems.

The first example of the direct system known to the writer was the circulation of brine from ice and salt through a system of pipe coils near the ceiling of the cold storage chambers. This system is still in general use

in many small plants throughout the country, and it is doubtless vastly superior to the indirect ice system, as it does away with the moisture and slop incident to the meltage of ice above the chambers, and insures dry, pure air, a factor of the utmost importance to the successful preservation of exposed perishable goods held in storage for any length of time. The direct system may be considered under several headings, depending upon the construction and location of the surfaces in the chambers; *i. e.*, whether exposed, or open circulation, or confined circulation of the refrigerant, direct ammonia expansion or brine circulation (including chloride of calcium) as the agents; but as the confined circulation will cover both the direct and the brine systems, and as the preference is given—justly, in the writer's opinion—to the confined systems, I will, for the time, drop the exposed system and treat only the confined system, as applied to both the direct and the indirect systems of refrigeration.

My observations of the adopted systems of confined circulation have deduced the following conclusions as to the relative amounts of surfaces required for successful refrigeration of well insulated cold storage chambers. For brine circulation: From one lineal foot of 1-inch pipe to each ten cubic feet of space to be cooled to a temperature of 32° to 34° F., to one lineal foot of 1-inch pipe to each five cubic

feet of space to be cooled to a temperature of 0° to 5° F., for sharp freezers. For direct ammonia expansion: From one lineal foot of 2-inch pipe to each forty cubic feet of space to be cooled to a temperature of 32° to 34° F., to one lineal foot of 2-inch pipe to twenty cubic feet of space to be cooled to a temperature of 0° to 5° F. for sharp freezers.

On the face of the above statement there is an apparent discrepancy, when the relative surfaces of the two sizes of pipe are taken into consideration, and also when the differing temperatures of brine and ammonia are considered (refer to table of pipe surfaces, page 130, "Compend of Mechanical Refrigeration"); yet the figures given will closely approach the actual practice. I shall later comment on the subject of pipe surfaces in connection with another branch of refrigerating apparatus, and will, therefore, allow practice to stand as the basis for my treatise on the cooling surface subject; but I will confidently assert that for practical cooling surfaces, equal surface being allowed, the best and most economical results will be obtained with the smaller diameter of pipe, with either brine or direct ammonia, conditions of operation of the balance of the refrigerating plant being assumed to be favorable to either pipe. Years since, the writer had occasion to make an extended investigation of the brine circulation system, for the purpose of collecting evidence to be

used in a threatened infringement suit. Notes taken at the time have been preserved, and extracts from the same have set the writer and some of his friends on the road to improvement on many occasions.

From notes on a large brewery in Ohio, I would call the attention of the reader to primitive construction in the line of confined brine circulation. The machine, brine tank, condenser, brine pump and balance of the refrigerating apparatus proper were of an old standard construction, more than amply large enough to easily perform the duty of refrigerating the brewery in question. The construction of the cellars was good, fully up to the standard at that time, but the piping of the cellars was such that the successful refrigeration of the brewery was simply impossible. To illustrate: One cellar, containing in the aggregate about 39,000 cubic feet of space, was piped, by actual measure, with 3,285 feet of 1-inch pipe *in one continuous coil*, connected to the main feed and return pipes only at each end of cellar. The coil was frosted less than one-third of its length, and the balance showed black and dripping with condensation of moisture from the air of the cellar. Every cellar in the brewery was piped in a similar manner. Two or three small cellars, containing from 400 to 600 feet of pipe each, showed coils all nicely frosted and temperatures as low as the brewer could reasonably ask for. A sugges-

tion was offered to the engineer that he divide the coils in the large room into sections of ten or twelve coils, and connect each section to the mains by means of manifolds, or headers. He replied that he was there to run the plant, not to fix it up; that it did not belong to the brewery, as it had never been accepted and probably never would be. He was running the machine at a ruinous speed, as he said, "just to find out how much he could get out of the old thing." He is still engineering—pumping sewage on a city political job, and has found his proper level.

Some time later the writer met the builder of the plant and told him about the pipe work. He assured me that he had never visited the brewery, having intrusted the work to a man whom he had reason to suppose knew his business, as he was a practical steam fitter and engineer. My friend thanked me for the information I had given him, and in token of his appreciation gave me a cigar, with which I secured the undisputed possession of the entire smoking compartment of a Pullman the same evening. Later on, my friend wrote me that the pipe work had been corrected, the plant accepted, and that he had got his "dough" out of it.

Three hundred to 400 feet of 1-inch pipe is the amount usually considered good practice for brine circulating coils. I have seen, however, 600 feet used successfully with low tem-

perature brine. Many contractors are now giving preference to 1¼-inch pipe on account of its being more rigid than the inch pipe. Instances occur where larger pipe is used, even as large as 6-inch, but the small pipe is the favorite all around.

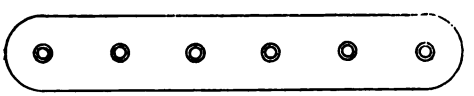
For direct ammonia expansion there seems to be no particular established length of coils. Where 2-inch pipe is used, the writer has noted a variance of from 800 to 2,000 feet to the coil, and even at the latter figure it would seem that the limit has not been reached. In the '70's Theo. Krausch experimented with direct ammonia; and I have heard it said that he remarked that "he did not believe he had money enough to purchase sufficient pipe to make a coil that anhydrous ammonia would not frost to its full length, if connected with suitable receiver and pump." I am inclined to think Mr. Krausch was right, and that it would take the resources of a Vanderbilt or a Gould to build such a coil, if liberally supplied with liquid and exhausted with a large pump.

CHAPTER II.

LOCATION OF PIPING.

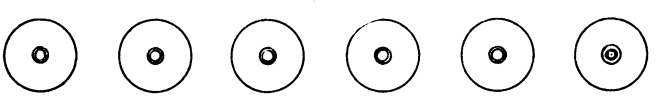
SO much for circulation. Now let us consider the construction and location of surfaces. As evidenced by the experience of the Ohio brewer referred to in the previous chapter, the early construction of cooling surfaces were devoid of the conveniences now considered absolute essentials to a thorough and satisfactory job of pipe work. Then the universal custom was to suspend the pipe coils as near the ceiling of the cooling chamber as possible or convenient, spacing the pipes about six inches apart from center to center, and, as far as possible, covering the entire length and breadth of the ceiling with pipe coils; and the zenith of a refrigerating engineer's ambition was attained when sufficient frost accumulated on the coils to close up the spaces between the pipes, and produce a continuous unbroken sheet of heavy frost all over the upper lines of the chamber. This system cooled the chambers and kept them cold—often colder than was necessary or desirable, and I regret to note that some refrigerating engineers yet cling to this undesirable practice, possibly on

the theory that it is best to let well enough alone. The evolution of refrigeration and consequent sharp competition in the trade have driven the thoughtful contractor to devise means to accomplish the same or better results at a less expense, and the ceiling piping has been spread to greater centers, saving pipe and securing better results by means of actually increased cooling surfaces and in the improved circulation of the air in the chamber. A reference to the sketch below will illustrate the point discussed.



6" Center to center of pipes.

Six 1-inch pipes frosted to a thickness of six inches, exposing a surface of 78.85 square inches for each inch of length.

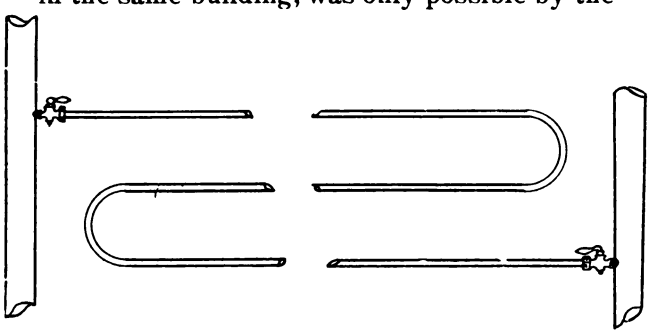


9" Center to center of pipes.

Six 1-inch pipes frosted to a thickness of six inches, exposing a surface of 112.09 square inches for each inch of length, and affording sufficient space for circulation of air between the surfaces. To the practical engineer further comment on this point would seem un-

necessary; but to the inexperienced I would say, spread your surfaces well apart, no matter what refrigerant you are using.

Early practice offered no means of cutting out an individual coil from the circulation without shutting off the entire chamber. It was a case of everything or nothing, and the regulation of the temperature in one chamber, as against the temperature in other chambers in the same building, was only possible by the



partial closing of the main valves to the manifolds, or headers, of a given chamber, or the opening of doors or windows when the temperature ran too low. The first alternative invariably resulted in dripping from part of the coils, at least; the second was only a makeshift at best. As the field of mechanical refrigeration broadened out and entered the premises of general produce storage houses, the necessity for controlling the temperatures of various chambers to a degree and at widely differing temperatures became apparent, and

this recognition led to many experiments and not a few expensive failures. The problem was solved when the ordinary steam circulating coil, with a shut-off at each end, was adopted for refrigerating surface coils, and the accepted brine circulation practice of to-day is illustrated by the foregoing sketch.

The best practice locates the return manifold at a point above the feed manifold, so as to insure that the circulation pipes are at all times full of brine, to prevent trapping of air in the pipes and consequent internal corrosion of the pipes. For the same reason, it is desirable that the feed and return mains should be constructed on the stand pipe system, extending above the highest circulation coils and dropping down to receive the circulation connections. Many complaints of brine pipes corroding out in a comparatively short time can be traced to a failure to observe this necessary feature in the original construction of the circulating system.

To my mind, for direct refrigeration the flat coil system near the ceiling of the chamber (but sufficiently removed from the ceiling to afford a free circulation of air above the coils) is the most desirable and practical system for general use. This system certainly insures more even temperature throughout the entire chamber than is possible with any other direct system. I appreciate that conditions are to be met with which make it desirable that the

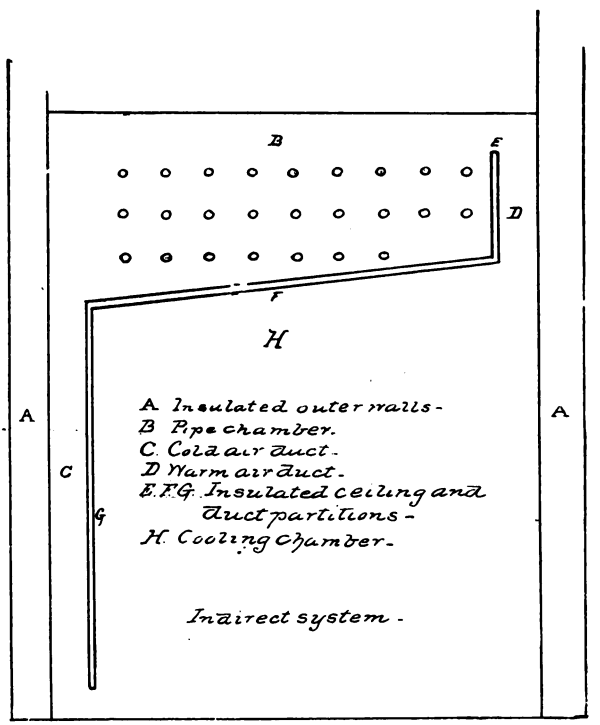
coils should not be suspended above certain lines of merchandise held in storage, as, for instance, where possible dripping of melting frost from the pipes would damage the goods. In such cases I would not recommend taking any chances with flat ceiling coils, but would group the pipes in box coils above convenient passages between the goods, or would arrange the coils around the walls of the chamber, as might best suit the conditions to be met. With both the box coil and the wall coil systems, the objectionable feature frequently complained of is sweating ceilings. The moisture in the air condenses on the ceiling, and very often drips down on the goods below.

Natural laws should be given due consideration in the construction of a cold storage plant. It is an established fact that cold air is heavier than warm air, consequently cold air will fall to the lowest level, and force the lighter warm air to the highest confined limits. Gravity attracts falling bodies, when not obstructed, in straight lines. Cold air, when not obstructed, will fall in direct lines from refrigerated surfaces, forcing up counter-currents of warm air to replace the falling currents of cold air. Diverting the natural direction of either current necessarily opposes natural laws; in consequence, good practice should suggest the advisability of utilizing natural forces to the fullest extent. Placing drip floors under cooling surfaces will divert nat-

ural currents and will deter results, possibly not dangerously but certainly appreciably. In some instances where the writer has been called upon to correct an unaccountable and unsatisfactory condition in a certain cooling chamber, the simple removal of an obstructing drip floor from beneath the cooling surfaces has promptly removed the source of annoyance. This feature is particularly noticeable as the frequent cause of sweating ceilings, a condition that is rarely met with where flat ceiling coils are adopted.

For chambers where danger from dripping must be guarded against, the indirect system of cooling surface is sometimes applied with good results. This system is necessarily a diverted current system, and the reader may be expected to take issue with me for recommending the indirect system immediately following my own assertions in relation to natural laws and the utilizing of the same. My excuse must be that where conditions exist making it impracticable to apply the best methods, then we must accept the inevitable and cast about us for the means to apply methods that will accomplish our purposes in the manner most suitable to all surroundings. In a few words, the indirect system may best be described as a pipe chamber directly over the cooling chamber and connected to the cooling chamber by one or more cold air ducts, leading from the lowest point in the pipe chamber to the lowest

point in the cooling chamber, and one or more warm air ducts leading from the highest point in the cooling chamber to a point above the cooling surfaces in the pipe chamber. To illustrate, I submit the sketch below:



In this construction it is essential that the cold air ducts and the ceiling below the pipe chamber should be thoroughly insulated, otherwise condensation of the moisture in

the air will be deposited on the wall and ceiling, especially when the chambers are opened for the reception and discharge of goods. When well insulated, such chambers will keep dry and sweet, and the feature of danger from dripping of melting frost from the pipes is effectually removed. Several eminent constructing engineers claim the honor of having invented this indirect system, but in the humble opinion of the writer it approaches so near to the lines of the old indirect ice refrigeration constructions as to be worthy only of the term adaptation rather than invention. It's older than I am; and although creeping along in years, I'm still "The Boy."

One of the many humbugs that have been and are constantly being introduced in the refrigerating field and in the opinion of many practical refrigerating engineers the most glaring humbug of all refrigerating humbugs up to the present, is the radiating disc so frequently used in connection with direct ammonia expansion pipe surfaces by some of the leading builders of refrigerating plants. The disc has come into such general use that ordinarily it is accepted by purchasers of refrigerating plants without so much as a passing thought regarding its practical utility. Jones has it on the pipes in his brewery; Smith's cold storage piping is covered with beautiful white discs, and they improve the appearance

of the piping. It must be all right or Jones and Smith would have discovered the fault and the discs would have been thrown out. How many times every refrigerating engineer and salesman, who has given the disc subject the consideration it deserves, has had to combat an argument similar to the above can only be guessed, not stated in figures. Too frequently Brown is not open to conviction and will have just what Jones and Smith have. The salesman wants to do the honest thing by his customer and offers to show the facts by comparative figuring; but you must not oppose the pet convictions of a possible customer or you will run a chance of losing his contract. The disc is a seductively appearing ornament, which is its only claim to recognition by a careful, thoughtful refrigerating engineer.

Let us compare some cold (not refrigerated) facts concerning the disc and pipe surfaces. The usual disc is about twelve inches in diameter, $\frac{3}{16}$-inch thick, tapering to $\frac{3}{8}$-inch thick toward the center, supported on the pipe by a boss, or flanged hub, about 1 inch to $1\frac{1}{4}$ inches wide. The surface of a 12-inch disc would be 113 square inches, minus hole in center for a 2-inch pipe, say 4.43 square inches = 108.57 square inches, multiplied by 2 = 217.14 square inches of surface for both sides of the disc. The disc weighs about nine pounds, and sells at from twenty-five to thirty cents each (for convenience, say aver-

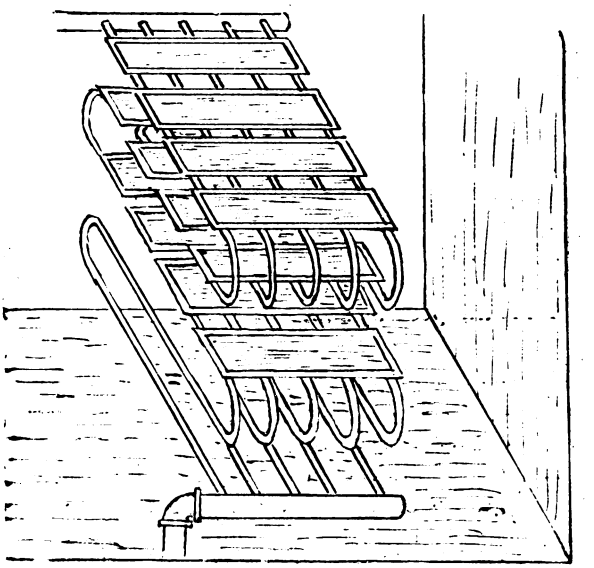
aging twenty-seven cents, or three cents per pound. Now, the actual outside diameter of a 2-inch pipe is 2.375 inches, circumference 7.46 inches, multiplied by 12 = 89.5 square inches, the surface of one foot of 2-inch pipe. Dividing the surface of a disc, 217.14 square inches, by the surface of one foot of 2-inch pipe, 89.5 square inches, we find that it takes 2.42 feet of 2-inch pipe to equal in surface the surface of one disc, and this on a basis of pipe and disc bare (free from frost). Now 2-inch pipe is worth about eight cents per foot, in large quantities; 2.42 feet × 8 = 19.36 cents for surface of pipe equal to surface of disc, 7.64 cents in favor of the pipe surface against the disc surface in cost of material.

And this is by no means the worse feature of the argument. We have compared the entire disc surface with equal surface of pipe, when actually such comparison has no place either in theory or practice, for the reason that all the heat radiated to the refrigerating agent through the medium of the disc must finally be transmitted through the intervening pipe surface; and as the disc bears on the pipe on a surface of about 1¼ inches wide by the line of circumference of the pipe (1.25 × 7.46 inches, or say about 9.32 square inches, of actual transmitting surface), our comparative surface calculations must finally resolve themselves down to the question of pipe surface as a finality, and the disc, pretty as it is, will

never find hanging room on piping for refrigerating surfaces in a plant owned by or operated under the control of a man who will stop to consider the subject long enough to arrive at a practical conclusion. The writer does not stand on a pedestal of virtue under the claim of never having sold a disc. I've sold car loads of discs, and may sell more of them to some of the Browns who must have them in their business; but I'll confess that I never bought a disc in my life, and I'll swear I never will buy one (unless I can sell it at a profit to some Brown who must have it). When constructing cooling surfaces, do it with good iron pipe, and put in enough pipe to give ample surface, but don't cover your piping with frills and ruffles. There's nothing practical in mechanical style.

Piping in freezing chambers varies in form, size and general construction fully as much as in other branches of refrigeration. In many cases the usual methods of hanging pipes from the ceilings are adopted, merely increasing the surfaces to double, or thereabouts, the amount usually applied to rooms for temperatures of from 32° to 34° F. In some cases wooden racks, fitted with hooks for hanging the materials which are to be frozen upon, are built around the pipe coils which are suspended from the walls, posts or ceilings. Some engineers prefer rigid box coils, from four to eight pipes high and three to six pipes wide,

located directly above the floors of the chambers. The coils are spread apart sufficiently vertically to serve as racks on which to place trays, or pans, containing the material to be frozen (see sketch). The trays, or pans, are filled in the packing or storage rooms and



taken into the freezing rooms on large warehouse trucks, to avoid opening and closing doors as much as possible. As one rack is filled with fresh material another rack of frozen material is loaded on to the truck and carted off to the packing and storage rooms. It is rare indeed to see a freezing room filled

with material much more than five feet from the floor. Warehouse men soon appreciate that the colder temperatures are found nearest to the floor and are quick to take advantage of this feature. In consequence of my own observations of this branch of refrigeration, I am led to the conclusion that a considerable waste space is refrigerated in freezing chambers unnecessarily, particularly so in cases where the material to be frozen is not excessively large in bulk. The hanging of carcasses or even quarters of beef necessarily requires a considerable height of the freezing chamber, and in such cases only is it necessary to refrigerate the entire chamber. My own preference of construction of freezers (approved of and adopted by some of my engineer friends) is the box coil system, near the floors, dividing the chamber into several compartments and inclosing each individual box coil in a thoroughly insulated compartment provided with doors on the sides for loading and discharging the freezers, and each individual freezer separated and isolated from its fellow in the same chamber. In this manner a freezer may be constructed against two of the walls with a double compartment freezer in the center of the chamber, leaving two convenient aisles to reach the fronts of the freezers to load and unload them, each freezer entirely independent of the others and consequently not affected by the opening and closing of its

fellow, and all of the freezers properly ventilated from the top into the common chamber. The chamber proper may or may not be refrigerated, as the engineer may determine. The main pipes will do considerable in this direction if connected to the coils at the far end of the chamber.

This system will be found economical and satisfactory for freezing small materials rapidly and nicely. It is not patented. The idea originated in a combination of a core oven and a bake shop cooling rack, a combination that calls to my mind the fact that quite a number of refrigerating inventions indorsed by "Uncle Sam's" patent office can be traced directly back to equally modest origins. Some other fellow originated the ideas and the cool refrigerating engineer collected them together in combination, applied them to cooling, and patented the "invention." This is the key note of the success of the up-to-date refrigerating engineer: he seizes and applies every good thing he meets that is applicable to his line of business, and if he is square it's ten to one he will tell you where he got the idea of that new (to you) good thing he has recently applied. Any first-class patent attorney could chase nine-tenths of the refrigerating patents off the face of the earth in a court of record trial.

The most desirable material for refrigerating surfaces is iron pipe. Several years' experience with steel pipe have proven that

the steel pipe will not last as long as the iron pipe. Especially is this fact noticeable in connection with brine piping. The writer put up iron pipe for brine circulation upwards of fifteen years since, that is to-day in constant use and to all intents and purposes appears to be as good as when first connected up. Steel pipe put up in a similar manner as regards connection and circulation has given out and become absolutely useless in from three to four years. I am aware of several instances where steel pipe has given entire satisfaction and can point to a few examples within my own personal experience; but the general average has been bad, so much so that I feel warranted in strongly recommending a preference for the iron pipe even at greater cost. The best is always the cheapest in the long run; and the best is none too good for use in every department and detail of refrigeration construction.

For return bends the practical engineer gives preference to extra strong iron pipe, bent to a radius of one-half the distance he wishes to locate his pipes from center to center. The pipe bends are threaded and connected with couplings on each end. The ordinary coupling usually furnished with random length pipe by the manufacturers should be taken off and disposed of, as they are of little or no value in connection with a first-class job of pipe work. They are too short and too light, and many times are tapped

straight through — a feature that almost precludes the possibility of making and maintaining tight joints, as the pipe is threaded taper and a taper threaded pipe will not make a first-class joint in a straight tapped coupling. I would, therefore, advise the reader to avoid the ordinary coupling, and would recommend in lieu thereof the extra long barrel form of chambered coupling, tapped taper from both ends, with a shoulder left between the threads in the center of the couplings. With such couplings tight joints can be depended upon for either brine or ammonia gas circulation. Such couplings are listed by almost all of the leading pipe and fitting manufacturers under such names as "patent coupling," "sleeve coupling," "hydraulic coupling," "special sleeve," etc., and while their cost is considerably more than for the ordinary coupling, their comparative value is far above the difference in cost.

Many engineers who have served their early time in shops where steam fitting is a part of the business, frequently make use of the right-and-left coupling for closing connections. This is a bad practice, and should never be used in connection with any part of a refrigerating plant; the uncertainty of making a joint on two pipes at once, and having an equal bearing when connected on both pipes, will condemn the right-and-left connection. The element of uncertainty should

always be avoided and never adopted on the plea of convenience. Any good flange union (or box union, for that matter) is preferable to a right-and-left connection and should be given the preference every time, unless something still better is at hand.

Every circulation coil should be provided with a valve, or cock (preferably, in ammonia circulation, with a valve), at each end of the coil to enable the cutting out of the coil at any time for examination and repair or to lessen the refrigerating surface in cases where the temperature falls below the point desired.

All mains and manifolds should be connected with flanged joints at convenient intervals, to enable the engineer to disconnect at any desired point in the system for the purpose of making any emergent repairs or to add further desired connections, without necessitating the breaking of a fitting or chiseling off a pipe to accomplish the purpose.

Mains, manifolds and circulation pipes should be kept well painted with good waterproof iron paint to prevent external corrosion when free from frost. Most engineers clean and paint all piping every winter, at least. Such is considered good practice. The writer has been asked many times: "What can we do to avoid this annual painting nuisance?" And the only solution which can be offered of the coundrum is, put in galvanized pipe. I want to say, however, that if the pipe is to be

used for direct ammonia expansion it must not be galvanized inside. One of the writer's errors of youth was to recommend (unintentionally of any wrong) the use of galvanized pipe in connection with a certain direct expansion plant in a room where it was necessary to cut off the circulation at times, and the engineer complained of rust when shut off. Following my advice, the engineer bought some galvanized pipe—the ordinary, commercial galvanized pipe, galvanized inside and outside. After the circulation was turned on that room again, there was an increase in condensing pressure on that plant that nearly scared the engineer to death (and incidentally kept "The Boy" and some of his friends guessing several hours before the difficulty was located). Permanent gases resulted from the decomposition of the zinc used in galvanizing the inside of the pipe when brought in contact with the ammonia; the pumps discharged the permanent gases into the condenser, and the gauge told the story. I gained some valuable experience blowing off permanent gases on that plant. So did some other fellows whom I called in to assist me. Not one of us will ever forget it, nor will we ever put up any more pipe, galvanized on the inside, for direct expansion. Any pipe manufacturer will furnish pipe, galvanized on the outside only, on special orders; and that is the kind of galvannized pipe to use for direct ammonia expansion.

CHAPTER III.

THE CONSTRUCTION AND PIPING OF BRINE TANKS.

LEAVING the investigation of chill room cooling surfaces, our next step would naturally lead us into a research of the source, or fountain head, of the refrigerating agent; and following the mains from the chill rooms, we would be led directly to the brine tank, if the plant should be a brine plant, or to the compressors, if the plant should be a direct ammonia expansion plant. Let us consider the plant as a brine plant, and look closely into the construction of the stored up refrigerating agent and its surroundings.

Arriving at the tank, we find it, in the majority of plants, ornamentally cased in with finished and oiled woodwork, and covered up with insulated covers, completely concealing its construction and effectually blocking our investigations. We perchance climb up on top of the woodwork, and raising one of the covers attempt to discover the contents. It is all dark within; but as we become accustomed to the darkness, we dimly perceive that the tank is full, or nearly so, with water. Dipping out a sample, we find it salt—very salt,

water; and as we are not provided with diving suits, we conclude not to drop down into the tank, and we hunt up and question the engineer on the construction of the tank, the form and size of coils it contains, construction and connection of manifolds, and—what not? Too frequently the engineer has never seen the brine out of the tank; and many times if he has seen the coils dry, he either cannot or will not give a satisfactory description of the contents of the tank. And it is possibly excusable in him, too, as many skeletons of mechanical hopes and many Waterloos of refrigerating experiments are sunk out of sight (but still in use) in many a brine tank in this country.

Well, as we can't dive into every tank we see, we shall be obliged to depend on the word of those who have been in the tanks and seen all that is in them and kept a record of what they have seen and are willing to tell us what the record shows. I am willing to "show my hand," and will tell you what my own experience has been. First, let me say that I have no desire to criticise any other man's work, nor to be at all personal in my remarks on my own observations. I will merely state facts, and make such comments on "things as I have found them," as will, in my opinion, enable any engineer with common sense to avoid being misled.

Brine tanks have been built almost without regard to any accepted theory or practice; and

tanks can be found that have been constructed to fit a given space rather than with a view to economy and practicability. There are round tanks, square tanks, deep tanks, shallow tanks, broak tanks, narrow tanks; and I remember one instance of a "well hole tank." It has been said that "variety is the spice of life," and if this is to be accepted as a fact, the statement could truthfully be enlarged upon and applied to the cold art by saying that brine tank variety is the cayenne pepper of refrigeration. No two builders adopt the same standards in either tanks or evaporating surfaces. The round tank is admissibly the cheapest tank in first cost that can be built, for the reason that it requires less bracing than any other form of tank. The pressure of contents being equal on the circumference at all points at a given height, it is, therefore, only necessary that the plates should be constructed of a sufficient thickness of metal to withstand the pressure of the contents of the tank and the seams properly made, and the result will be a good tank. Round tanks are, however, the poorest refrigerating tanks extant; for the reason that the coils must be made to conform to the shape of the tank.

And right here this practical feature must be given the consideration it deserves. Spiral coils are the meanest coils made to handle, to clean, to repair—to do anything with. It is next to impossible to adapt a set of spiral coils

to a round brine tank and secure area openings from the coils of the same aggregate area as the area of the ammonia suction pipe (a feature of the utmost importance to economical results) and at the same time space the coils to the best advantage in the tank. Any one who has been obliged to handle, clean and repair spiral coils out of a round brine tank will advise you to avoid them.

Round tanks will not locate nicely into corners of rooms. They leave a waste, useless space in the corner. They expose more surface to radiation proportionate to their contents than any other form of tank. No round tanks and spiral coils for "yours truly." Square and oblong tanks will locate nicely in wall corners, and when insulated from the walls expose but two sides to radiation. Coils can be put in so as to occupy the space to the best advantage, and area openings to the coils can readily be accommodated to the area of the suction pipe. It will cost more money for square and oblong tanks than for round tanks, but you are getting value for your money, and that is the only real consideration in a refrigerating trade.

To give the reader an idea of first-class tank construction, I submit a copy * of an old standard tank specification, in blank; and the

* SPECIFICATION:

....

.....

.....

.....

Please name us your lowest price on the following speci-

writer has on file a list of over 500 tanks, running in value above $250,000, that were built on similar specifications, and there was little or no trouble with any of them. The bands around the body of the tanks were made of two 3×3-inch angle irons, with a flat plate ½×6 inches riveted between, the angles bent around the corners of the tanks, and the flat plates lapped and riveted together at the corners. On tanks six feet deep, one band was put on; tanks eight feet deep, two bands; tanks twelve feet deep, three bands; all spaced

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fied tank delivered.....viz.:.....Open top tank of.....in. iron. .....long x.....wide x..... ft. deep, with.....in. x..... in. angle iron around top, and .....bands around body of tank made each from.....

Tank to have.....brace rods the..... ..ft. way, each made from.....in..... .....iron, and fastened at each end by..... ....

Tank to be further braced by..... ..... All as per accompanying sketch.

In the construction of the above work, only first-class material and workmanship shall be employed. All corners of tank to be made by bending to.....radius, or made with..... in. angle iron, as may be preferred by contractor. No split calking to be allowed about tank . All rivets to be hand driven, no snapping allowed, and all sheets to be well laid up. No lead to be used in making joints. Tank to be filled with water when completed, to test same, and guaranteed water tight, and free from leaks of any description, and not to bulge when filled with water. Tank to be painted after testing with first-class..... ....

Tank to be subject to our inspection and approval, and any work found defective either in material or workmanship to be satisfactorily rectified without any additional expense to us.

Tank to be delivered, as above specified, complete, and finished by.....18...., without fail.

Awaiting your earliest reply, we remain, yours very truly, ...

so as to equalize the strain on the tank plates from top to bottom. Tanks six feet deep were made of 3-16-inch plate; eight to twelve feet deep, ¼-inch plate; fifteen feet and deeper, ⅜-inch plate on the bottom course, 5-16-inch on the center course, and ¼-inch or 3-16-inch on the top course. Long tanks were braced crosswise with stay rods of 1-inch or 1¼-inch square iron fastened to the inside of the tank by a pair of 5×3-inch angle irons running vertically, the rods having eyes or rings turned on the ends which were held in place by pins, or bolts, through the two angle irons on each end. Generally turnbuckles were put in the center of the rods to tighten up the rods by. All tanks were painted with two coats of waterproof paint inside and outside. Some of those old tanks will still be doing business when you and I are "pushing clouds." They were put up to stay.

Speaking about putting up a tank, there is only one way to put up a refrigerating tank, and that is the right way; and the right way is to dig down until you strike blue clay, hard pan, or other solid footing, and run up good hard brick or stone piers to the floor level; lay good 2-inch plank over the piers; then two courses of first-class waterproof insulating paper, one course crossing the other; then rough boards over the paper; then 2×12 joists on edge, 12-inch centers, strongly battened up at both ends and in the center, filling the

space between the joists with granulated cork, rammed in tight; then rough boards over the joists, and two courses of paper again over the boards, and finish over the paper with good, clear dressed and matched flooring. When the tank is in position on such a foundation, dam up all around the tank, and pour hot pitch under the entire bottom of the tank. Such a foundation will pay for itself in a year by keeping heat from entering the tank from below. When the tank is in permanent position, set up joists or furring strips around the exposed surfaces; nail on rough boards, keeping about one foot away from the face of the tank, beginning at the bottom, and as fast as the boards are put on, fill in the space between the tank and the boards with granulated cork, well rammed in. Where the tank is placed in a corner, keep it out about a foot from both walls, and fill in with cork between the tank and the walls, as above described. Over the rough boards put on two courses of paper, and finish over the paper with good flooring. Don't waste money on fancy finished lagging; save it to put in some other detail that will give you adequate returns. Make a good set of well insulated covers in sections small enough to handle easily, and you are ready for your coils.

And you had better stop right here, and do some hard thinking before you put a single coil in that tank; because between the evapo-

rating surfaces in the tank and the ability of your compressors to relieve the coils of the ammonia they are capable of evaporating and expanding, rests the principal efficiency and economy of the plant. Here I will submit a table that I have compiled in a hurry. It will convey intelligently in a small space what would necessitate many pages of description. The figures are from actual experiences in practice; the proportions are the result of thought and care, and you can rest assured they are safe.

TABLE OF BRINE TANKS AND COILS.

Capacity in Tons Refrigeration.	Length of Tank in Feet.	Width of Tank in Feet.	Height of Tank in Feet.	Thickness of Iron, Inches.	Cubic Feet of Brine when Filled within Six In. of Top.	Number of Coils.	Pipes High.	Feet Long.	Size of Pipe.	Total Feet of Coilage in Brine Tank.	Feet of Coilage per Ton of Refrigeration.
25 tons.	16	13	8½	¼	1,664	38	12	11	1	5,016	200
35 " ..	20	13	8½	¼	2,080	46	12	11	1	6,072	173.5
50 " ..	20	13	11	¼	2,730	46	16	11	1	8,188	163.4
75 " ..	22	15	15	⅜	4,785	52	20	13	1	13,963	186
										4)	722.9
Av'rage per ton	...	...	.....	...	.....	..	..	..	..	.....	180.7

NOTE.—While the 50-ton brine tank contained the lowest average evaporating surface per ton of rated capacity, yet it has proven ample, when operated in connection with good compressors, as was demonstrated by Prof. Denton's reports of tests, in which a 50-ton Consolidated machine, in connection with a brine tank and evaporating surfaces of the above dimensions, actually performed a refrigerating duty equal to the melting of seventy-five tons of ice in twenty-four hours.

Now a few words in relation to the size of pipe for evaporating surfaces: First, no matter

what size of pipe you decide upon, don't let some chump try to influence you to put common pipe into a brine tank. Buy and pay for extra strong iron pipe, and you will save many dollars in the long run. Don't cover up any skeletons in the depths of your brine tank. Good material is trouble enough to take care of, without hunting for more by sinking poor material under brine.

Every first-class refrigerating engineer in this country who is honest in what he says will recommend small pipe in preference to large pipe for all heat transmitting surfaces. You ask why? I will answer: Because he knows that the advantage, when volume of contents and surface are compared, is largely in favor of the small pipe. I would advise every owner of a refrigerating plant, and every engineer operating one, to purchase a copy of the "Compend of Mechanical Refrigeration," published by H. S. Rich & Co., and familiarize himself with the two tables on pages 129 and 130 of same on the subject, the "Dimensions of Pipes." These two tables alone are of more value to the interested student of refrigeration than the price of the book ten times over. For convenience, I borrow the following data from the subject as treated by Prof. Siebel in the above book:

"One running foot of 2-inch pipe is equal to 1.44 feet of 1¼-inch pipe, and 1.8 feet of 1-inch pipe as regards surface."

From the common pipe table I deduce the following:

"Length of pipe containing one cubic foot, 2-inch pipe, 42.36 feet; 1¼-inch pipe, 96.25 feet; 1-inch pipe, 166 feet."

Examine the above closely. The 2-inch pipe has 44 per cent more surface than the 1¼-inch pipe, and 80 per cent more surface than the 1-inch pipe. The content of the 2-inch pipe is 2.27 times that of the 1¼-inch pipe, and nearly four times that of the 1-inch pipe; and when considering extra strong pipe, while the surfaces are exactly the same as in the case of the common pipe, the volume is still greater proportionally. The only rational conclusion, therefore, which can be deduced from the comparison is that the large pipe does not, comparatively speaking, expose sufficient surface to transmit the heat its content is capable of absorbing; and to make use of an equivalent comparison in practice, I would assume that a man who would use large pipe for heat transmitting surface in connection with a refrigerating plant could reasonably be expected to give preference to an old style flue boiler over a modern tubular or a water tube boiler for making steam. The principle involved is identical in either case.

CHAPTER IV.

BRINE TANK COILS.

HAVING decided upon the size of piping to be used for evaporating and expansion surfaces and the proportionate number of feet of such pipe to be used per ton of refrigerating capacity, the next step is to determine the most practical and economical, as well as most convenient manner, of locating the coils in position in the brine tank and connecting up the coils to the liquid and suction systems, with reference to securing the best results.

It is conceded that there is a wide difference of opinion existing among the shining lights of the profession in relation to the form of evaporating coils and the manner of connecting up the same. The diversity of opinion as to form (or shape) is aptly illustrated by the advertisements of various coil manufacturers; and while these advertisements do not cover the entire range of opinion, they will serve to illustrate the fact that the authorities do not all agree on any one particular form or shape of coil.

I have paused a few moments here to light my pipe and look over an old memorandum

book containing some sketches of evaporating coils, from the long-ago and up-to-date, illustrating other people's as well as my own exploded theories and fair successes. I find spiral coils, elliptic coils, flat coils, with screwed or flanged return bends, welded and bent flat coils, Baudelot coils and a number of other shapes, most of which are unworthy of mention; but a note, without date, made some time before 1884, will bear repeating, if not interest. The memorandum reads: "Of all the coils heretofore sketched and described, the welded and bent flat coil is the most practical, for the following reasons: (1) Experience has proved that it can be built more economically than any other form of coil; (2) it will load to better advantage on a car for shipment than any other form; (3) it is the most convenient form of coil for handling in and out of a tank; (4) being devoid of screwed joints from end to end of coil, the liability of leakage in a tank and consequent delay in erecting a plant is reduced to the minimum."

I made that memorandum more than twelve years ago, and at this time I have no reason to warrant me in changing the verdict. In my opinion the welded and bent flat coil should be given the preference, all surroundings being considered.

Three systems of connecting up evaporating coils have been and are now used in brine and freezing tanks, and each one of the three

has its strong advocates among the practical refrigerating engineers and experts of the art. For convenience I will divide the systems and treat them under captions which, through use rather than suitability, have become their names, viz.: "Top Feed," "Bottom Feed," "Top Feed and Bottom Expansion."

The first of these systems which came under my notice was the top feed system, which consisted of a liquid manifold, made of ½-inch, ¾-inch or 1-inch extra strong pipe, with a regulating, or, as we used to call it, an evaporating valve for each individual evaporating coil. This liquid manifold was connected to the top end of the coils by means of ammonia unions, nipples being screwed and soldered into the valves for the union connection to the coils. In the center of the liquid manifold a tee was placed to receive the pipe connection from the liquid receiver. The gas, or suction, manifold, made of extra strong pipe, usually the same size as the suction pipe to the compressors, and provided with a nipple for union connection to the coils, was located and connected to the bottom ends of all of the evaporating coils. This manifold was fitted with a tee in the center, to receive the suction pipe connection, and also with a *purger pipe* connection on the bottom side of the manifold, leading out through the tank (and provided with a packing gland, or flange, to prevent brine from leaking out of the tank), and a

purge valve outside of the tank. The evaporating valves (sometimes called distributer valves) were regulated (as far as possible by experimenting with them) to give an equal distribution of liquid to each individual coil, and the evaporating or back pressure was then controlled entirely by means of a main regulating valve on the liquid pipe outside of the tank.

At times it would be found necessary to "blow out," "flush" or "purge" the liquid manifold, owing to oil from the compressors coming over with the liquid and lodging in the valves, where it would freeze and clog up the openings to the coils and prevent the accomplishment of satisfactory results in consequence. The purging process was one of the drawbacks to the engineer's joy. Many an old time engineer will remember his "salt-junk fat hands" after a session of purging; and as he reads these lines he will smile, because it's all different now. The purging process consisted of closing every valve on the liquid manifold; and while the machine was running slow, open one at a time wide, allowing a rush of liquid through the valve to melt out and carry the oil down through the coil to the bottom manifold; then close the valve tight, and repeat the operation on the next valve, and keep on repeating until every valve was purged; and then open and regulate the valves one at a time. And it was hours, sometimes days, before the exact, delicate combination

was working satisfactorily again. Some way or another, the brine would always manage to get up over the valve wheels, and the purging was done three times out of five with the bare hand immersed in *cold brine*,—ye gods! how cold! Many's the time your humble servant would willingly have affirmed that it was 30° below zero when it actually stood 14° above; but two or three hours in 14° brine would tempt a saint to lie.

After purging the liquid manifold, the purger on the bottom manifold would be tried, and the usual result was from two ounces to half a pint or more of a yellowish or milky liquid that was good stuff to get out of a plant as soon as possible. It would saponify and stick so tight to a glass or can that it could not be washed out; and stink—well, there are only two things I can compare that stink to: one is the aroma of a morgue; the other is the odor from a near relative of the original: the smell from the residue of cheap, gas house ammonia offered at a few cents less than the market and worth dollars less than nothing (as the purchaser finds, to his sorrow, when too late).

Right here I want to digress from the subject in hand long enough to pay a deserved tribute to the energy and honesty of the original anhydrous ammonia manufacturers of this country. When I first became interested in the subject of refrigeration and ice making, we made our own ammonia from the aqua.

We made it with a crude apparatus, and it cost us a barrel of money to produce an inferior article. Larkin & Shaffer, of St. Louis, Mo., now a strong factor in the National Ammonia Co., were the first to recognize the inevitable gigantic strides the then new industry would make in this country, and they were shrewd enough to take prompt advantage of the opportunity presented. At enormous expense (as compared with later competition prices) they installed a plant for the manufacture of anhydrous ammonia, and they made ANHYDROUS AMMONIA, such as you don't get from gas house liquors. It was made from the sulphate, the best sulphate they could buy, and you could not carry a sample of their anhydrous around, corked up in a pop bottle on a hot day (as the writer did some cheap goods last summer). Early successes in refrigeration were due as much to Larkin & Shaffer's ammonia as they were to the machines. They were the pioneers in the business, who chopped out and cleared the land for the present progress. The writer has paid their bills for ammonia at better than $1 a pound, and received 100 cents on the dollar in value for every dollar he ever paid them. Some day the consumer will learn that he can't get something for nothing (as the nigger puts it), and when he does, *anhydrous* ammonia will not sell for anything like twenty cents per pound. *Good* ammonia at $1 a pound is *cheap* to the consumer when com-

pared with the truck the writer lately tested at twenty-two cents per pound. And I am not saying whose ammonia it was, either.

TOP FEED.—To resume: The top feed system has its advantages, and it also has its objectionable features. The advantage will be appreciated by every engineer who will stop and think a little. The arguments in favor of top feed are the following:

First.—Liquid fed in the top of a coil will naturally seek the lowest point, and in doing so (if not fed into the coil too fast) will spread itself out to the furthest limit, *exposing the greatest possible surface to evaporation.* If you have observed the water in a boiler when it was being carried too high, you have noticed that the boiler does not "steam" well. Why? Because the *evaporating surface* decreases as the water line runs up the circle of the shell, and you have not room or surface enough to promptly liberate the steam from the water as rapidly as it can be formed. What is the difference, in principle, between a boiler and a brine tank? None; both are mediums, or conveyances, for the transmission of heat; one to utilize the heat, the other to dispose of it. In either case the main object is to *transmit* the heat in the most economical manner. Study the question from this standpoint and you will reach a sensible conclusion. When the liquid reaches the end of the first run of pipe in the evaporating coil, it trickles down the bend to

the second run. Gravity assists the operation. It spreads out again, once more increasing the evaporating surface, and so on down the coil, until (if properly regulated) the liquid has changed to a *gas* and expands fully before reaching the suction manifold.

Second.—Observation has demonstrated to all of us that there is a difference of several degrees between the temperature of the brine at the top and that at the bottom of a brine tank. Especially is this noticeable in a deep tank. Why? Because the return brine from the chill rooms is necessarily warmer than the discharge brine from the brine pump, otherwise the circulation of the brine would not have accomplished any purpose. Where there is the most heat, there will of necessity be the most ready evaporation. If you don't believe this assertion, close the dampers of your boiler for a few moments and watch the steam pressure run down, and be convinced by practical observation. The top of the brine is where we will find the most heat. Let us therefore take advantage of this fact and evaporate the ammonia at the point where we have the greatest facility for rapid evaporation; in other words, where we can evaporate the greatest amount of ammonia. As fast as the brine is chilled it will go down to the bottom of the tank, and as the gas expands in the coils it will chill the brine and help gravity to push it along down.

The disadvantages of the top feed are: (1) If too much liquid is fed to one or more coils, it will run down into the suction manifold without evaporating, or expanding, to its utmost; and it is then sucked to the compressors without having accomplished the purpose for which it was fed into the coils. (2) If oil is carried over with the liquid to the coils, it will gradually coat the interior of the coils with a soapy lining that will partially prevent ready transmission of heat, and consequently retard results. The saponified oil is hard to get rid of without removing the coils from the tank and steaming or pickling them out thoroughly.

BOTTOM FEED.—This system consists of reversing the manifolds as above described, *i. e.*, placing the liquid manifold at the bottom of the coils (usually, however, dispensing with the valves and making the manifold of large pipe) and the gas or suction manifold at the top of the coils. The liquid and suction connections are the same in either system (always providing the bottom manifold with a purger pipe and valve). The advantages claimed for the bottom feed system are:

First.—That the liquid enters the tank at the coldest point, evaporating only such an amount of liquid as the surrounding brine temperature calls for; and as the gas expands and travels up through the coils, the interchange of temperature from the brine to the gas is accomplished in the most theoretically correct

manner, and the gas will leave the tank at nearer the same temperature of the warmest brine than is possible with any other system.

Second.—That the liquid being fed at the bottom of the coils will absolutely prevent the lodging of oil in the coils; on the theory that when the oil becomes saponified by exposure to low temperature, it will not travel up hill, but will lodge in the bottom manifold, keeping the coils clean and in the best condition for maximum transmission of heat.

Third.—That with a large liquid manifold in the bottom of the brine tank the major portion of the liquid may be carried to the manifold and the manifold actually used as the liquid receiver; where the liquid will lie cold in consequence of its own evaporation.

Fourth.—That the necessity of constant regulation of liquid valves is reduced to a minimum; for the reason that if too much liquid is fed into the manifold, it will merely lie there cold without evaporating, and will not prove detrimental to the operation of the plant.

The disadvantages are: (1) The reducing of evaporating surface to a minimum. You cannot crowd a bottom feed system one inch beyond the evaporation called for by the temperature of the coldest brine in the tank.

(2) If the liquid valve is opened too much, and the lower pipes of the coils are flooded with liquid, in consequence the evaporating surface will be confined to the area of the coils

and not to the length of the pipes. To illustrate the point, we will say that the lower pipes are filled full of liquid up to the center of the first bend. We would have the area of one bend multiplied by the number of coils in the tank, as the total evaporating surface. Or a better illustration would be to take a 1-inch pipe, 10 feet long, and fill it half full of liquid. When it is placed in a vertical position, the liquid would expose only $\frac{1864}{10000}$ square inches for evaporating surface. Now lay the pipe down horizontally, and the liquid would spread out until it exposed 120 square inches for evaporating surface. The comparative merits of the top feed and the bottom feed systems, as regards the point of evaporating surface exposed, are aptly illustrated by the above suggested experiment. And that one illustration is sufficient to condemn the bottom feed system.

TOP FEED AND BOTTOM EXPANSION.—This system is a combination of the best elements of the two systems above described. Each alternate coil in a tank is connected to a liquid manifold (provided with regulating valves) at the top of the tank, and the ammonia is evaporated downward through one-half of the coils in the tank. All of the coils in the tank are connected to a large bottom manifold (which might be called an equalizing expansion manifold), and the gas is returned up through the second half of the coils to a gas suction mani-

fold at the top of the tank, located behind and a little above the liquid manifold. The suction manifold is provided with a tee for connecting to the suction pipe to the compressors.

The advantages of this system are: (1) The evaporation of the liquid at the top of the tank, where the brine is the warmest; (2) double the travel of the gas that is possible with either of the first two systems; (3) if one coil gets too much liquid and another not enough, the surplus is trapped in the bottom, or equalizing, manifold, and the return coils will be given an equal distribution of gas to the gas suction manifold.

This system has all of the advantages and none of the disadvantages of the other two systems, with the one exception of the oil clogging feature, and later I will give a remedy that will cure that evil effectually. The only disadvantage I have ever heard mentioned of the system was called to my attention by a particular friend of mine, whose ancestors sprang from the tribe of Judah. He said: "Grashoos, my poy, dose costs doo mooch!" But I put "dose" in his tank, just the same, and he is glad of it.

Now when connecting your coils to the suction manifold, pay strict attention to one point, and it will save you lots of trouble: Put in coils enough so that their combined area will equal the area of the suction pipe to the compressors. If you want more coils than

your suction pipe area will allow, put in as many coils as you need, and choke off the area openings from the coils to the suction manifold by using double extra strong nipples, reducers or reducing flange unions until the combined areas of all of your openings to the coils equals the area of the suction pipe and no more. Use good bolted flanged ammonia unions to connect up the coils to the manifolds, so that you can take the joints apart readily and quickly whenever it becomes necessary to remove the coils from the tank. The best is the cheapest in the long run everywhere about an ammonia plant.

CHAPTER V.

THE LIQUID TRAP AND DRYER.

TWO necessary details of construction in an ice making or refrigerating plant are neglected entirely by the majority of manufacturers and engineers. Where the engineer neglects them it is generally owing to prejudice. As a rule, the manufacturer avoids them merely to save expense in the cost of the plant. I refer to the liquid trap and the dryer. I will venture the assertion that no unprejudiced engineer who has given the details mentioned a fair and impartial trial will consent to their being overlooked in the construction of any plant that is to be operated under his supervision. These details are essential, whether the plant be operated for brine circulation or for direct ammonia expansion; and in the humble opinion of the writer, acquired in the rough school of experience, no plant is complete without them—not that it is necessary that they should be kept in constant use, but that they may be in place and connected up ready for use when the occasion for their use arrives, as arise it does, sooner or later, on every plant.

The liquid trap consists of a chamber, usually constructed of a short piece of large-diameter pipe, with heads welded in both ends, or with ends capped, as may be most convenient. The trap is placed on the liquid pipe, as close as possible to the brine or freezing tank or to the direct expansion coils in the chill room or rooms. An expansion valve is placed on the inlet of the trap and an ordinary ammonia valve on the outlet of the trap. Usually a by-pass is made in the liquid pipe so that when necessary the liquid may be carried around the trap and the trap disconnected for cleaning. A purge valve is located in the bottom of the trap to allow of discharging or purging the contents of the trap when in use, if so desired.

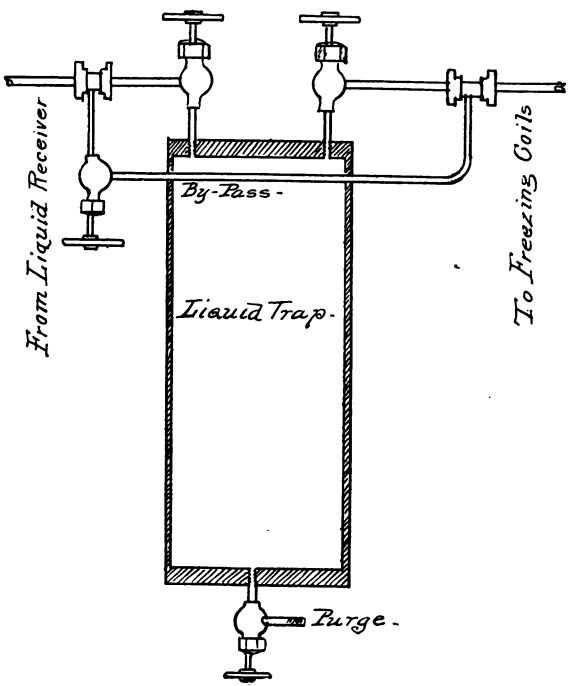
To operate the trap, the by-pass connections are closed and the outlet valve from the trap (leading to the coil connections) is opened wide. The evaporating pressure is controlled entirely by the regulation of the expansion valve at the inlet of the trap. Should the ammonia be contaminated by oil, weak liquid or other undesirable matter, excepting such matter only as liquids equally as volatile as anhydrous ammonia, the deleterious substances will lodge in the trap, and will not be carried over into the expansion coils. The writer knows of several instances where liquid traps have freed ammonia systems of weak liquors sufficiently in a few hours to obtain

satisfactory results, where before the trap was introduced the plant was unable to perform the functions required of it by at least 20 per cent.

The objectionable feature in connection with the trap is the fact that the ammonia is evaporated outside of the tank, and there is in consequence a loss of a portion of the available heat-absorbing properties of the ammonia. It is impracticable to place the trap within the tank, first, on account of its being inaccessible when so placed; and, second, on account of the difficulty that would be experienced in purging the trap of accumulated oil in case the trap was surrounded with cold brine. The trap could not readily be warmed up and the oil purged out in a liquid state; the cold brine would keep the oil in a heavy, saponified condition and render purging a difficult, if not an impossible, operation. The trap should therefore be located where it can be easily got at and, when necessary, where it can be warmed up, outside of the tank. When used in connection with direct expansion piping in chill rooms, the trap can be located in any convenient position in the room or in the hallways outside of the rooms, as may be most desirable and practical.

As to the size of the trap, there is no rule which needs following. I have seen traps used all the way from four inches to ten inches in diameter and from one foot to four feet in length. See that you get sufficient area in

the head to allow for drilling and tapping the inlet and outlet connections, and make the length of the trap to suit your own fancy. The less surface exposed, however, the less will be the loss due to radiation of heat into the gas.



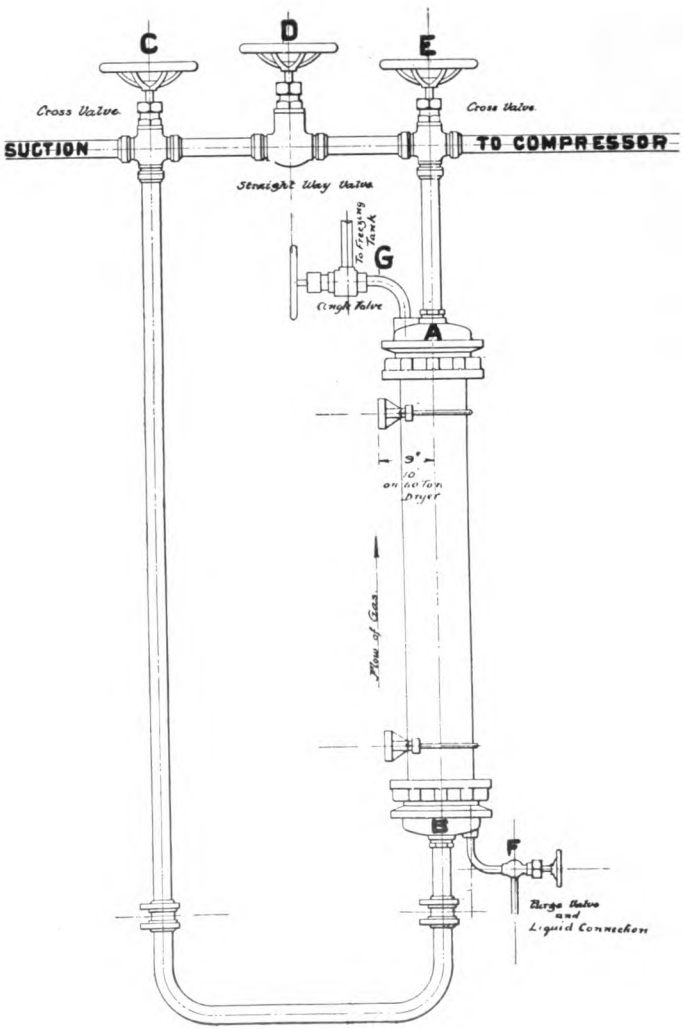
THE LIQUID TRAP.

The accompanying diagram is a sketch of the trap, which is usually made of two feet of 6-inch or 8-inch pipe, with heads welded in each end, or with caps screwed and soldered on, and fitted with connections the same size

as the liquid line. The reader will fully understand the trap and its construction from the sketch.

Try the purge valve of the trap frequently when the trap is in use. When the trap is clean and dry inside, it will frost all over, which condition will also obtain when the trap contains weak liquor; but when the trap contains saponified oil, it will tell you the fact at a glance. The oil seems to be a non-conductor, or insulator (as many of my readers have ascertained when it worked into their expansion coils), and the frost leaves the trap when its interior is coated with saponified oil. You can see just how much oil there is in the trap by the frost line indicator. Sometimes, however, you will find a milky, or yellowish, weak liquor will purge from the trap when it is frosted all over. I explain this by the same reasoning that has demonstrated that a sample of ammonia tested in a glass or can will frost every portion of the surface of the receptacle below the liquid line, whether the ammonia be good or bad. A liquid that will not freeze will transmit heat much more readily than a frozen liquid.

The dryer can be described as a trap on the suction pipe, or connected in such a manner that the suction gas can be passed through it when necessary; said trap to be provided with removable heads for filling and discharging, and the trap to be filled with a moisture-



SKETCH FOR AMMONIA DRYER.

absorbing substance, such as lime (unslaked) or caustic soda. The diagram which appears on the opposite page shows the construction of the dryer, which may be described as follows:

The body of the dryer consists of a length of 10-inch or 12-inch pipe from four feet to eight feet long, with removable heads *A* and *B*. Some users fill the body, or shell, with a system of perforated trays made of wire screen or perforated sheet steel, about one foot apart, held up by flat iron stands, or feet. The trays are put in from the top of the shell, and as each tray is put in it is filled up with fresh, dry lime or caustic soda. Others use only one tray at the bottom, and fill up the dryer shell to the top with lime or caustic soda. The suction pipe enters the bottom of the dryer, and extends up about six inches to form a trap and prevent particles of lime from falling down into the pipe.

To operate the trap, close the straightway valve *D* and open the cross-valves *C* and *E*, and the gas will pass down to the bottom of the dryer, up through the body of lime and out at the top to the compressor. To pump out the trap, close the cross-valve *C*, leaving the cross-valve *E* open, and pump until the gauge shows a good vacuum; then close the cross-valve *E* and open the straightway valve *D* and the plant can be kept in operation while the dryer is being discharged and replen-

ished. To discharge the dryer, unbolt the lower head and swing it around out of the way, and allow the contents of the shell to drop out. Then replace the lower head, and refill the dryer from the upper end.

The angle valve *F* answers a double purpose: (1) It can be used as a purger for blowing off moisture from the bottom of the trap; (2) the charging connection can be made to this valve; and when charging the plant with ammonia, the gas can be passed directly through the body of the lime or the caustic soda to the compressor, or it can be passed through the body of lime or caustic soda to the expansion coils by making a cross-connection from the valve *G* to the expansion coils.

The majority of the anhydrous ammonia manufacturers fill their dryers (every one of them uses a dryer) with lime; a few still use caustic soda at times, when some crank like the writer insists on it. The lime is the safest material for general use, as the caustic soda is not a pleasant material to get into the eyes, nostrils and mouth. The caustic soda should, therefore, if used at all, be used carefully, guarding against the dust flying around promiscuously.

The criticism has been made that were the dryer filled to the top with burned lime as an absorbent of moisture, it would gradually assume the condition of a fine powder, occupying

a much greater volume than the original lime; and my correspondent says: "Without being assured to the contrary by one who has practical experience in the premises, I should expect that in the course of a little time the dryer shell would be completely choked with the voluminous powder of slaked lime, preventing practically the passage of ammonia altogether. For this reason I always supposed that a dryer with burned lime should have a horizontal position, and that ample space should be left for the lime to expand and for the ammonia to pass overhead. Caustic soda I would expect to behave somewhat differently. It assumes the condition of a liquid when it has absorbed a certain amount of moisture, and in the course of time it would collect as a liquid on the bottom of the dryer, and by and by it would overflow into the ammonia pipe entering at the bottom of the dryer even if the same projects six inches into the dryer. These are conjectures based on theoretical considerations, and, as I said before, are probably refuted by practical experience with the device in question."

Now as to this, I can say that in the majority of instances my own experience coincides with the foregoing theory, so far as the lime is concerned. I have found that the lime will disintegrate to a fine powder; but as the lime is usually put into the dryer in large lumps, not packed closely, I have never found that the

powder or lime dust occupied as great a space in the dryer as the original charge of fresh lump lime took up. Undoubtedly, as the lime disintegrates, the powder sifts down to the lowest point if not obstructed by the trays recommended in my remarks on the construction of the apparatus. In a few instances I have found the powdered lime in more or less a condition of paste, but I am safe in asserting that in these instances the plant had become fouled with moisture or that the dryers were too small and were operated too long before recharging with fresh lime.

I am not inclined to favor the writer's suggestion to make the dryer horizontal instead of vertical, for the reason that the horizontal dryer would afford an opportunity for the lime to settle down on the bottom of the dryer, creating a space above the body of lime, over which the ammonia gas could pass without coming in direct contact with the body of lime. My preference is to make the dryer vertical, and either fit it up with the system of trays containing the lime, or fill the shell with large lumps of lime and pass the gas directly through the body of lime, thus bringing all of the gas in direct contact with the lime, and not depending upon the affinity of the lime to attract moisture to draw the moisture out of the gas indirectly. I have never seen a dryer choked with the powdered lime sufficient to prevent free passage of gas

through the dryer, but it is possible that others have met with this condition of affairs.

My experience with caustic soda as an absorbent has been limited, owing to the fact that I found it a dangerous material to handle, liable to get into the eyes and cause them to become badly irritated; and I have had other experiences with it that have caused me to recommend that it be handled with extreme caution. When the dry dust of caustic soda gets on a man who is perspiring freely he is forcibly reminded of the fact that he has some business of importance to transact elsewhere, and if the dust happens to wander up the leg of his trousers, that business assumes an entirely personal nature and demands his immediate and undivided attention.

I have never seen caustic soda assume the liquid state, probably because I have never seen it used in a plant sufficiently permeated with moisture to bring about such a condition. I have seen it discharged from a dryer in a more or less pasty condition, and have also seen (and felt) it when it came out dry and dusty. If it will assume the liquid state, I would by all means advise the user to leave it in the dryer until it liquefies, and then purge it off from time to time as the liquid accumulates in the bottom of the dryer, as I am sure that the liquid would be a much easier commodity to dispose of than the dust has proven

to be. That caustic soda is a good material to extract the moisture from ammonia gas has been demonstrated to my entire satisfaction, but I most emphatically recommend the users to exercise care in handling the stuff.

The dryer, judiciously used, will remove any moisture from the ammonia in a plant, whether the moisture got there with the initial charge or was due to carelessness of the engineer. You are aware that if you put a cylinder of ammonia into a plant without testing it and satisfying yourself that it is all right, you cannot fairly claim that its faults, if you find it faulty later on, are due to the original quality of the ammonia, and ask the manufacturer to make it good. You find after a time that your ammonia does not work exactly right; it doesn't seem to respond to your requirements. You sit down and write to the manufacturer, and lose valuable time trying to convince him that it is all his fault. Don't do it. You are the best man on earth to help yourself out of a scrape. If you are "in the hole" (an expression coined particularly for ice machine operatives with a bad charge in a plant) go to work yourself to get out, and the chances are ten to one that you will smile before you could have got an answer to your kick to the ammonia manufacturer.

When there is anything wrong with a plant, there is a reason for it. Find that reason your-

self, and you will be in a position to rectify the wrong and guard against its recurrence. If you happen to get left on a purchase, don't "holler" and advertise yourself a "sucker." Get all you can out of your purchase, and charge the balance to profit and loss; and remember the offspring of Satan who dared to "do" an ice machine man. Put up a dryer of your own, and it will pay you its cost many times over, if you get caught in hot weather. A friend of mine acted on this same suggestion some time since, and he took one hundred and twenty pounds of useless moisture out of a large plant in a very short time (he weighed his lime in and out of the dryer); and he wouldn't be without a dryer again, no matter where he got his ammonia. He doesn't claim that the moisture went into the plant with the ammonia, either. Some manufacturers claim to make their ammonia $99\frac{99}{100}$ per cent anhydrous. Give us a first-class dryer, and work the ammonia through it often enough, and we will hunt out that $\frac{1}{100}$ per cent of moisture before many moons. But the other fellow, who doesn't make better than 90 to 95 per cent pure goods, and claims 110 per cent pure, and the careless engineer, who pumps a vacuum on a leaky joint under water and sucks his system full of moisture, are the fellows whom the dryer will tell the toughest tales on and whose evils it will correct more economically to the owner of the plant than is possible by

the means of the only other alternative in such emergencies, viz., the purge valve, the sewer and a fresh charge of ammonia. I've "been there," and there's many another who could voice the same sentiments if called upon to confess.

The honest ammonia manufacturer will join me in recommending a dryer on every ice making and refrigerating plant in the country. It would cut down their troubles to a minimum in the line of answering many unreasonable kicks. The majority of them are anxious to furnish a good article, and would gladly welcome any device that will expose the shark in the trade, be he the unprincipled manufacturer or the careless engineer.

I have gone to greater length than I had intended in the treatment of the two details recommended, but I feel warranted in this, if by doing so I can accomplish my purpose in assisting my fellow-sufferer out of the rut that prejudice and penuriousness (of some manufacturers) would force and hold him in.

CHAPTER VI.

THE COMPRESSOR SIDE.

TO resume my theme where it was dropped in Chapter IV, I find the next step to be taken will lead to the suction pipe to the ammonia compressor. Of this connection there is little necessary to be said further than to admonish my reader to have the suction connection large—as large as possible; never less than 25 per cent of the diameter of the compressor, and as much larger as convenience of connections will permit. The object of the suction pipe is to convey to the compressor from the evaporating and expansion surfaces the spent gas by the shortest and easiest route and to deliver the same to the compressor at as near the pressure obtaining in the coils as possible. In all my experience with this class of machinery, I have never seen a suction pipe that was, in my opinion, too large; on the other hand, I have met with many that were far too small for practical results, and were in consequence retarding the economical action of the compressor.

A correctly designed and constructed compressor will discharge readily all of the gas

that can be crowded into it, and as the results depend entirely upon the amount of gas handled by the compressor (suitable evaporating and expanding surfaces being assumed), it would appear to be policy to make the delivery of the gas to the compressor as free and easy as possible; especially so when it is taken into consideration that we have not got high pressure on the expansion, or suction, side to assist in filling the compressor rapidly and fully, and also that in most cases we have to overcome the tension of a suction valve spring before the compressor will begin to fill. The suction pipe and the suction valve should, therefore, be given as much area as possible.

Now we arrive at the compressor—the lungs of our circulating system, the vital force on which the utility of our plant depends, the one part of a plant on which there is, unfortunately, more variance of opinion among the authorities than on all other details of a plant combined.

Ammonia compressors may be divided into two classes: single-acting and double-acting. These classes subdivide into two forms: horizontal and vertical. All ammonia compressors, including the compounds, are covered in the above classifications. In the selection of an ammonia compressor, dollars and cents (in first cost) should be given but secondary consideration as compared with utility, economy and service. In the present field of strife in

competition, I regret to note that the dollar is given the entire consideration by both parties to the deal, and as a result value is, in the majority of instances, shelved, to be discovered in the later on.

Personally I have long since ceased to pit the truth against the dollar. There is no common ground on which the contestants can meet, and when they do meet truth will prevail and the dollar be a quantity of the past. Competition has driven every manufacturer in the country to the dollar side of the question, and it is now a matter of "How many dollars have you got? We'll cut your garment accordingly." (Quotation from a late actual proposition.)

I recommend you to buy and pay for single-acting vertical compressors—single-acting, because two single-acting compressors will displace more gas than one double-acting compressor, diameter, stroke and piston speed being identical in both cases; vertical, because a vertical compressor is perfectly balanced, and will outwear any horizontal compressor, all conditions being equal. The seating of the valve of a vertical compressor can be made so that the wear is equal at all points on the valve and seat; the piston bears equally on all points of the circumference of the cylinder in a vertical compressor. Weight of piston and rod causes no unequal bearing. Compressing on the upper end of the stroke heats only the

upper end of the cylinder and upper side of the piston. The space below the piston is constantly filled and refilled with cool or cold gas, which assists in maintaining the walls of the cylinder and the piston reasonably cool. There is necessity for care in the matter of clearance only on the upper end of the piston stroke, and as the wear is necessarily all down, the clearance in a single-acting vertical compressor can be safely reduced to the minimum without fear of knocking out a cylinder head. Dry gas or saturated gas can be handled by a single-acting vertical compressor. Examples of each are in constant operation right here in Chicago, and can be seen any time, day or night. No allowance need be made for piston rod displacement in a single-acting compressor, as no compressing of gas is done in the piston rod (or stuffing box) end of the compressor. It is only necessary to pack the stuffing box of a single-acting compressor against the evaporating or expansion pressure of the ammonia (generally amounting to about 10 per cent to 15 per cent of the condensing pressure). Lubrication of a vertical compressor is equal all around the cylinder and piston. In fact, the only objections that can be offered against the single-acting compressor are first cost and friction of two pumps as against one when compared with double-acting compressors; and from cards in my possession, I feel warranted in the as-

sertion that the friction argument will not hold good, as the packing of the stuffing box of a double-acting compressor, against condensing pressure (six to ten times the pressure necessary to pack against with a single-acting compressor), will more than counterbalance the friction of the single-acting compressor.

Twenty years of close observation of single-acting compressors as against double-acting compressors is the basis of my opinion of the question in making a recommendation, and the twenty-year-old compressors of the single-acting type are still running. The double-acting compressor is the father of the wet, or saturated, gas theory, and the paternity is due to "making a virtue of necessity." A double-acting compressor will not handle dry gas in competition with a single-acting compressor. Freeze up your double-acting compressor and you will *heat up your piston and cylinder* to such an extent that the cylinder will not fill with gas from the expansion coils. The heat of the cylinder and piston will expand the gas as it enters the cylinder, excluding a full charge from entering the cylinder.

Now, I may be wrong, and if I am wrong, I am open to conviction. I, however, don't put any stock in the saturated gas theory. As I understand by "saturated" gas, it is intended to mean a gas that has not fully performed its functions before it enters the compressors, and it is utilized to *cool the compressors.* It can-

not cool the compressors without expanding; if it expands in the compressors, it necessarily excludes a proportionate amount of gas from entering the compressors at the same temperature and pressure that the gas leaves the expansion coils. In other words, this process curtails the capacity of the compressors for handling gas at the temperature and pressure it leaves the expansion coils—makes a smaller compressor, as it were.

The coil cooling theory in an ammonia compressor is bad enough; this is worse. Again, I have always been led to believe that the more the gas was expanded in the expansion coils, the greater amount of heat it would carry with it from the tank or chill room, and the nearer the temperature of the gas equalized with the temperature of the surrounding brine or atmosphere, as the case might be, the nearer would be the approach to absolute perfection of duty performed by the evaporation and expansion of the gas; that the place to perform the refrigeration was in the expansion coils and *not outside of the tank or chill rooms.*

If this theory is correct, then the saturated gas theory is all wrong; and the engineer who is pumping gas that has not performed its full functions before it enters the compressors, is doing something for nothing—wasting energy. I would as soon think of running an engine with "saturated" steam as of pumping "saturated" ammonia gas in a refrigerating machine. The economy would be equivalent in either case.

CHAPTER VII.

CONSTRUCTION OF AN AMMONIA COMPRESSOR.

THE construction of an ammonia compressor, whether single or double-acting, should be given serious consideration from every possible point of view before final decision is arrived at relative to its proportions and details. The assertion is frequently made that "the science of refrigeration is entirely covered by the three simple propositions: Adequate expansion surface, ample condensing surface, and sufficient compressor displacement." For the sake of the argument, I am willing to grant that these factors cover the root of the question; yet I must contend that all three factors can be constructed of ample proportions and still not meet the requirements of practicability and economy.

In my articles on the subjects of pipe surfaces and brine tank coils, I have endeavored to treat the subjects in a manner that would appeal to the *reason* of the engineer; and it shall be my purpose in speaking of the compressor to bring out plainly common sense ideas, based on experience and thought. I am no evangelist endeavoring to convert the

world to my way of thinking and believing, but I have sufficient confidence in the intelligence of the engineering profession to warrant me in the assertion that the average engineer will recognize the truth or falsity of a claim or theory *if he will take time to think over the proposition.*

The purpose of the ammonia compressor is to take in and discharge as much gas that has fully performed its function as its maximum cubical contents would represent. The engineer who is merely superficial or theoretical would be satisfied with a determination of the area of the cylinder multiplied by the length of stroke of the piston as a demonstration of displacement accomplished by the compressor, possibly with an allowance for apparent clearance deducted to give his observations the semblance of accuracy; but the careful, practical engineer will dive much deeper and will take into consideration all surrounding conditions, especially the conditions of practicability, economy of operation and accessibility of parts, before giving his opinion of the *value* of a compressor.

The reader must remember that competition has driven the manufacturer into the position of striving to build his machine at the least possible cost without detriment to the materials and workmanship employed. He therefore strives to accomplish the greatest possible displacement of compressor with the

least possible cost of construction. The result is the short-stroke, large-diameter pump, where years ago the long-stroke, small-diameter pump was the accepted practice. Which is the best pump for the purchaser and the engineer (granting both to have identical cubical capacity)?

The long-stroke, small-diameter pump is unquestionably the better of the two. Granting that clearance can be reduced to $\frac{1}{32}$-inch in either pump (and there are more pumps in operation that are allowed more than $\frac{1}{16}$-inch clearance than there are pumps working with a clearance of $\frac{1}{16}$-inch or less), let us make a comparison of say a 10-inch and a 12-inch-diameter pump, each with a clearance of $\frac{1}{32}$-inch, on the basis of a stroke for each piston that would call for an equivalent displacement in each cylinder, and see where we will come out on a day's operation. The 10-inch cylinder has an area of 78.54 square inches. The 12-inch pump has an area of 113 square inches. As the clearance is necessarily at the discharge end of the piston stroke, clearance must be calculated at *terminal* pressure; and this terminal pressure must be equalized with the initial pressure, at least, *before the suction valve will open to admit a fresh supply of gas at the original initial pressure.* If we figure on a basis of 150 pounds absolute terminal pressure and $\frac{1}{32}$-inch clearance to expand down to an initial pressure of say twenty

pounds, theoretically when the piston receded from the head double the distance, or to $\frac{1}{16}$-inch from the head, the pressure would be one-half of 150 pounds, or seventy-five pounds; when the piston reached $\frac{1}{8}$-inch from the head, the pressure would be one-half of seventy-five pounds, or 37.5 pounds. Again, doubling the distance of travel of the piston from the head, we find that the pressure of the gas at $\frac{1}{4}$-inch clearance would be 18.75 pounds, or about the point where the suction valve might be expected to open to receive a fresh supply of gas. Our supposed $\frac{1}{32}$-inch clearance is therefore actually $\frac{1}{4}$-inch clearance when calculated on the basis of initial pressure; and in calculating the capacity of an ammonia pump, initial pressure is the factor to be considered.

Now for our comparison. Our 10-inch pump, with an area of 78.54 square inches and a terminal clearance of $\frac{1}{32}$-inch, or an initial clearance of $\frac{1}{4}$-inch, loses one-fourth of 78.54 cubic inches, or 19.63 cubic inches of displacement for each stroke. Under precisely similar conditions the 12-inch pump would lose one-fourth of 113 cubic inches, or 28.25 cubic inches of displacement per stroke—a difference in favor of the small diameter pump of 8.62 cubic inches, or nearly 50 per cent greater loss for the large-diameter pump. Figuring each pump at sixty strokes per minute for twenty-four hours per day, we find the difference in favor of the small-diameter

pump to be represented by the following, viz.: $8.62 \times 60 \times 60 \times 24 = 744.768$ cubic inches of displacement, and this on one single-acting compressor working at the minimum of clearance. With two compressors (or one double-acting compressor), the total would be double the displacement above given; and as the actual clearance is often double that given as the basis of the calculation, it is apparent that the economy is with the small-diameter and long-stroke compressor, and against the large-diameter and short-stroke compressor.

The only disadvantage that can be claimed against small-diameter compressors is that they do not offer as easy facility for locating large area of valves as do the larger-diameter compressors, but it is only necessary to give consideration to this feature in the smaller sizes of compressors, as the larger sizes afford ample room for locating suitable valves in the heads.

The long-stroke compressor also covers a feature of desirable mechanical action that should recommend it to every engineer, viz.: Maximum piston speed at minimum revolutions of the crank shaft. Every thoughtful engineer recognizes that the principal factor of wear and tear on an engine or compressor is the dead point or point of change in the piston stroke, and the practical engineer gives the preference to the engine or compressor that accomplishes the maximum of piston speed at the minimum of change points.

Some manufacturers of ammonia compressors have designed pistons for compressors of conical or spherical form, probably to facilitate placing the valves in the compressor heads to advantage. This practice *increases clearance area*, and is therefore undesirable. Show the practical engineer a way to decrease clearance area below the actual area of the bore of the cylinder, and he will adopt the idea suggested; but a practical engineer will never favor *increasing clearance area.*

Other manufacturers recommend the injection of a limited amount of oil at each stroke of the piston "to do away with the clearance." The practical engineer *knows* this to be wrong, both in theory and practice. One of the laws of physics, taught in every primary school, is "that no two things can occupy the same space at the same time." On this basis every particle of oil injected into an ammonia compressor must necessarily exclude an equivalent particle of ammonia gas from entering the compressor, and the result is the conversion of a gas pump to an oil pump in exact proportion to the amount of oil injected. The engineer who has made close observations also knows that under compression oil will take up a considerable amount of ammonia gas, and when relieved of its pressure will again give off the gas rapidly. All compressors arranged to operate on the oil injection principle are constructed with generous clearances between

the piston and the heads. It would be dangerous to run them with the clearance reduced to a minimum, for the reason that no gas pump is designed with discharge-valve areas as large as those provided on pumps designed for pumping liquids, and without free discharge areas a liquid pump would be liable to knock out a head.

Liquids are not elastic; gases *are* elastic. When the piston of an ammonia compressor recedes from the head, if the clearance space be occupied by oil charged with ammonia gas, the instant the oil is released from pressure, that instant the oil will begin to give off the gas and the gas will begin to expand and exclude a proportionate amount of initial pressure gas from entering the cylinder. Given one compressor working without oil injection with the piston set to the minimum of safe clearance, and another compressor of exactly equivalent displacement working with oil injection, with the piston set at the clearance required for oil injection, both working at the same piston speed on the same size of ice making apparatus, and I'll stake my reputation that the scales will render a verdict in favor of *no oil injection.* And when the test of equivalent piston speed is decided, the no-oil-injection compressor can safely be speeded up from 20 to 25 per cent beyond the safe speed of the oil-injection compressor; and the verdict of scales will then forever damn oil in-

jection in an ammonia compressor. I believe in *rational* lubrication of compressors, but I reprehend *overdoing* the lubrication, and that is all "oil injection" amounts to.

The oil injection is sometimes called "oil sealing," to cover up its iniquity with a cloak of virtue. Any compressor that needs a flood of oil at each stroke to conceal its imperfections belongs in the scrap pile, and should be promptly replaced with a first-class gas pump. Put your oil pump in a pipe line and get an ammonia pump.

The piston of an ammonia compressor should be made so as to afford a good, broad bearing on the walls of the compressor. It should be turned an easy and yet a reasonably close fit to the cylinder, and be fitted with at least two snap rings, with a shoulder between the ring grooves. The rings should be turned larger in diameter than the bore of the cylinder, and then cut zig-zag, so as to make a lap joint where cut; then clamped at the joint so as to spring in nearly to the diameter of the cylinder, and should then be turned on the outside to the exact bore of the cylinder. Such rings when in place in the cylinder will spring out and hold close to the walls of the cylinder, and with slight lubrication will run easy and prevent leakage by the piston, at the same time reducing the wear to the minimum and requiring absolutely no adjustment.

Piston rods should be as large as convenience of the stuffing box will permit, and parallel all around. It is not absolutely necessary that a piston rod should be round. It *is* necessary that it should be straight and parallel on its surface, otherwise it will drag on the packing and leak through the stuffing box in spite of the greatest care and watching. Oval rods, if parallel, will work better than round rods when tapering or fluted.

If you have a good compressor, you can safely set the piston flush with the end of the cylinder, using a straight edge across the cylinder and screwing the piston out until it touches the straight edge. Then $\frac{1}{16}$-inch of packing between the cylinder and the head, with the head drawn home firm on the studs, will compress the packing until the clearance is about $\frac{1}{32}$-inch, which is getting it down fine.

Stuffing boxes need be no longer for ammonia gas than for a steam engine. With good rods and any first-class soft packing, or the better class of elastic-backed metallic packings, there is no reason why the stuffing box of a single-acting ammonia compressor should not be easily kept perfectly tight and free from leakage. With double-acting compressors, it is the practice to use double stuffing boxes, with an inside and an outside packing space and an oil circulating chamber between the two packings, through which a constant circulation of oil is pumped or run by press-

ure and gravity device. The oil is to keep the packing soft and prevent its heating the rod. The oil is also utilized, at times, to conceal the ammonia leakage from the inside stuffing box and carry the gas over to an oil cooler, which, by the way, is more of an ornament than a necessity in connection with an ammonia compressor; but it covers up the smell of escaping gas, and I shouldn't be surprised if that is all it was ever intended to accomplish.

Water jackets should be used on all compressors, whether single or double-acting, wet or dry compression. Any heat of compression that can be disposed of through the cylinder by means of a water jacket is surely more economical than to try to dispose of it at the expense of the capacity of the compressor, and every other means of disposing of this heat at the compressor is at the expense of compressor capacity.

Valve areas in an ammonia compressor should be at least equal to the area of the suction and discharge pipes. The valves should seat *flush with the heads* so as to leave *no clearance spaces around the valve chambers.* Clearance is what counts to the detriment of an ammonia compressor, whether it be produced by imperfect setting of the piston, by the space occupied by the piston rod (in double-acting compressors), or the unoccupied space around valve seats. The main factor in the construction of an ammonia compressor *is to reduce the*

clearance to a minimum, and when so reduced, use the compressor to pump the refrigerating agent, not wind, oil or anything else. *Heat* is what you are trying to dispose of. Get all the heat you possibly can carry with ammonia gas into the compressor, and aim to discharge all the heat you possibly can from the compressor. The ammonia gas is but the vehicle for carrying the heat. The compressor is but the power for moving the vehicle along.

Construct your compressors with a stroke at least two and one-half times (three times would be better) the diameter of the bore; give the piston ample bearing in the cylinder, and good wide snap rings; reduce clearance to a minimum; give ample suction and discharge area; use water jackets, and *pump gas.*

CHAPTER VIII.

OIL INJECTION.

WHEN treating this subject originally in *Ice and Refrigeration*, I was confronted by the following interrogatories by C. A. Lozano, M.E., a gentleman of learning and experience in the science of refrigeration, which, for the sake of completeness, I introduce here. Mr. Lozano said:

"1. Don't you know it to be a fact that oil is injected into ammonia compressors during the compression period, after the cylinder is fully charged, when the oil does not occupy any space to the exclusion of gas?

"2. Do you know how much ammonia gas at ordinary compressor temperatures and pressures will be given off by unit volume of the oil, before and during the period of suction?

"3. What is to hinder any competent engineer, having a well made compressor, arranged to operate on the oil injection principle, from running such compressor safely with the clearance between the piston and the heads reduced to $\frac{1}{16}$-inch and less?

"4. Is it not a fact that there are in operation hundreds of double-acting ammonia com-

pressors cooled internally by oil injection, and which, contrary to what 'The Boy' asserts to be 'the practice,' neither need nor have double stuffing-boxes?

"5. Are there no educated, competent machine designers, no competent mechanics, engaged in the manufacture of double-acting ammonia compressors cooled internally by injection of oil during the period of compression?"

At the risk of repeating something that has been before said, in the interest of further clearness, I here take up the questions in their order:

1. I do know it to be a fact that oil is injected into ammonia compressors during the compression period, but I take issue with the gentleman on the hypothesis of the balance of his question, viz., "After the cylinder is fully charged, when the oil does not occupy any space to the exclusion of gas."

He certainly does not claim that the oil is discharged with the gas from the cylinder at the same stroke when the injection occurs. He will, no doubt, admit that such portion of the oil as may be necessary to entirely fill up the clearance spaces will remain in the cylinder after the completion of the compression, regardless of the position of the piston at the time the injection is admitted to the cylinder.

I do not know of any process that would expose the oil to more complete permeation of

gas than is secured by the injection of the oil into the cylinder "during the compression period"; and if the oil that remains in the cylinder after the compression is completed is so permeated with gas at terminal pressure, I contend that the remaining oil will give off the gas and the gas will expand down to initial pressure before the suction valve will open to receive more gas into the cylinder on the suction stroke of the piston. It strikes the writer that it is a matter of very little consequence to the results *when* the injection occurs; the oil will be there beyond the completion of the compression, and the oil will contain as much gas at terminal pressure as it is capable of carrying, and that gas must expand down to initial pressure before *fresh* gas will be admitted to the cylinder. If this does not act to the exclusion of an equivalent amount of *fresh* initial pressure gas, the writer will back down from his position.

2. I do not *know* positively "how much ammonia gas, at ordinary compressor temperatures and pressures, will be given off by a unit volume of oil before and during the period of suction."

I have tried several times to demonstrate this proposition, but I must confess that my efforts were not crowned with the success I had anticipated; for the reason that so far I have only succeeded in securing *foam* from a compressor, at the terminal pressure, instead

of "a unit volume of oil." The stuff seems to churn up too much to admit of getting it out in its normal condition. My inquisitor, and any other experienced engineer in the business, will appreciate the difficulties to be surmounted in order to establish to a positive certainty the exact relationship between the gas and the oil *within the compressor.*

I have, however, secured samples of oil from the high pressure side of a plant, and made some (to me) interesting experiments with the same. I find that all oils do not act alike under high pressures of ammonia gas, an experience that has doubtless been reached by others; but from my own experiments I am of the opinion that a given volume of the average mineral oil will take up at least from three to four times its volume of ammonia gas under high pressure. To illustrate: A given quantity of oil under high pressure was injected direct from a high pressure oil interceptor under a piston into a small cylinder, and when the piston was released, it would rise to from three to four times the height occupied by the oil, the reverse side of the piston when released having merely atmospheric pressure on it. These experiments cannot, of course, be taken as a criterion of the conditions existing within the compressor, as under the conditions mentioned the oil was not in the same condition it would necessarily be in when in the compressor. The experi-

ments, however, demonstrate that the oil will take up a considerable quantity of gas, and that it will again give off the gas immediately on being released from pressure. Every ice machine engineer knows the condition of oil purged from a high pressure interceptor—how it will come out foaming and will fill a bucket to the top and give off gas until it finally settles down to a comparatively small volume. This fact alone demonstrates the claim that oil will take up gas under high pressure.

3. The hindrance a "competent engineer, having a well made compressor arranged to operate on the oil injection principle," seems to have "from running such compressor safely with the clearance between the piston and the heads reduced to one-sixteenth inch and less," would appear to be his desire to keep his machine in running order and hold his job.

Not long since I stood by while an erecting engineer, whom I know to be competent, was instructing a new man, and a good one too, in the art of operating an oil injection compressor. (I am not yet too old to learn, and, to tell the truth, I was "rubber-necking.") I heard that, "It is best to keep the pistons from three-sixteenths to one-quarter inch away from the heads, so as to avoid pounding the bearings." I have found pistons in this class of compressors fully three-eighths of an inch away from the heads, and I have yet to see a single in-

stance where the pistons were set within one-sixteenth of an inch from the heads in a compressor operating with oil injection.

A friend of mine a short time since tried oil injection in a compressor that was working close. He swore. I got an order for a large pump head. It's all right now, without "injection."

A prominent engineer, operating one of the largest refrigerating plants in the United States, who has under his charge both double-acting oil injection compressors and single-acting compressors without the oil injection, tells me that he "cannot run the oil injection compressors safely up to the piston speed at which he runs the other compressors; that the oil injection system is a source of constant annoyance to him, in that the oil carries minute particles of grit and scale over to the valves in spite of the most careful filtering and straining of the oil, resulting in the necessity of regrinding the valves three or four times in a season; and that the evaporating surfaces become contaminated with oil in spite of his most careful efforts to prevent the oil from working over; *in fine*, that he can discover no advantage from the injection of oil into ammonia compressors—rather a detriment than an advantage."

4. It is a fact that there are a large number of double-acting ammonia compressors, both with and without oil injection, working

with single stuffing boxes on the piston rods. There are, however, vastly larger numbers working *without* oil injection and *with* double stuffing boxes, and more of them are being built right along.

I have no apologies to offer for having made use of the term "practice" in this connection. Webster defines "practice" as "customary use, habit." The single stuffing box on the double-acting ammonia compressor is not "practice," nor has it been for many years.

5. There may be "educated, competent machine designers and competent mechanics engaged in the manufacture of double-acting ammonia compressors, cooled internally by injection of oil during the period of compression," but compared with the number of competent machine designers and competent mechanics in the ice machine business who are not engaged in building "oil pumps," their number is so small that it is not to be wondered at when they get lost in the shuffle. We hear of them sometimes, but more often we don't, and later on we won't.

That I advocate "the vertical, single-acting compressor, cooled externally by water jacketing, pumping gas superheated to the highest temperature (if it can be called properly superheating) consistent with the degree of cooling to be done," *is a fact*, as Mr. Lozano intimated elsewhere in his correspondence, and I do it consistently. I will bank on that

theory until I have seen it demonstrated that it is wrong. I learned it from better men in the business than any "oil pump" manufacturers, and it has stood the test of years of actual practice in the art. Many expert tests covering the widest range of conditions have been made by entirely disinterested experts on this design of compressor, and their reports have never been questioned, nor have equivalent results ever been reached with any other ammonia compressor.

These replies to the above interrogatories brought forth from Mr. Lozano the following remarks, which I have deemed best to incorporate here, as well as a further communication upon the same topic by Mr. Walter K. Edgar, an experienced and intelligent practical refrigerating machine engineer, both of whom have presented ideas worthy of the reader's consideration. Mr. Lozano, in substance, says:

"I have no interest whatever in common with any concern manufacturing oil-injected compressors, and I do not advocate oil injection or any other refrigerating machine nostrum; but I am conversant with all the practical advantages and drawbacks of oil injection in ammonia compressors, by dint of hard and long experience, and I assert nothing but what can easily be confirmed by plain facts if need be.

"1. 'The Boy' admits that he knows that oil is injected into ammonia compressors dur-

ing the compression period. That admission contradicts his former assertion, that every particle of oil injected into an ammonia compressor must necessarily exclude an equivalent particle of ammonia gas from entering the compressor. Of course, the oil is discharged with the gas from the cylinder at the same stroke when injection occurs, all but such portion of the oil as may be necessary to entirely fill the clearance spaces; and if that portion contains ammonia gas under discharge pressure, some of that gas may re-expand before and during the following suction period and act to the exclusion of fresh initial pressure gas; and if so, we should be able to see it in the indicator cards of the compressor.

"2. That the amount of gas given off by unit volume of oil before and during the period of suction may be very small, is proven by the indicator cards taken at the ordinary temperatures and pressures and under the conditions recommended by those who know how to handle oil-injected compressors to best advantage. Any number of such cards may be taken, in which even with a good magnifying glass one cannot discern any trace of re-expansion.

"3. If 'The Boy' wishes to see oil-injected compressors running safely with the clearance between the piston and the head reduced to one-sixteenth of an inch and less, I

shall be pleased to provide him with an introduction to parties who can show him a few.

"4. Quite right; but the fact is, nevertheless, that there are in operation hundreds of double-acting, oil-injected ammonia compressors which neither need nor have double stuffing boxes, although a well made double stuffing box is no detriment.

"I admit that oil injection in ammonia compressors, except at the rate of a few drops of oil per minute for lubrication only, is doomed to extinction; but I contend that the drawbacks of oil injection are different from anything which one would naturally infer from 'The Boy's' teaching. They are these: The sudden impact of the oil against the cylinder head and its sudden reaction against the piston at the end of every stroke, resulting in unnecessary stresses inevitably transmitted all the way down to the main bearings, making it very difficult, if not impossible, to operate an oil-injected machine without pounding and unnecessarily increasing the wear and tear on the machine. The frequent, if not general, troubles with oil going over to the expansion coils, either by failure of the complicated and expensive oil separating apparatus or by slight inattention of the operators, which in the long run is almost inevitable. The fact that displacement which is near enough to perfection is easily attainable without oil injection; and the fact that internal cooling of

an ammonia compressor by oil is little, if any, better than external cooling by water jacket, being neither low enough in temperature for anything approaching maximum efficiency, nor susceptible of being made lower. At lower temperatures the absorption of ammonia by the oil and the consequent re-expansion of ammonia in the cylinder before and during the period of suction increase at astonishingly high rates, as the compressor indicator cards will show. The heavy unnecessary expense of oil pumps or oil injectors and the mechanism used for actuating them, and oil coolers, oil connections, oil piping, oil fittings, oil cocks, etc., all high priced and calling for extra work of regulating and care-taking, from which the user does not derive equivalent return."

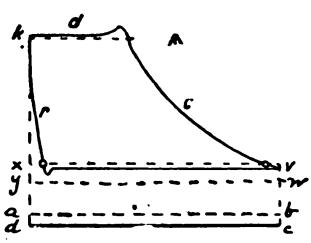
Mr. Edgar's views are as follows:

"Mr. C. A. Lozano advances some arguments in reply which should not, in my estimation, be allowed to go unchallenged, although they are in line with the theories advocated by a well known refrigerating machine company. While I have no doubts of 'The Boy's' abilities to defend the position he has assumed, the point that I take the liberty of taking exception to is one that he has only touched in a very superficial manner. I refer to the probable amount of re-evaporation of ammonia gas from the residual oil in the compressor after complete discharge has taken place.

"In his communication above, Mr. Lozano apparently lays considerable stress upon his thorough acquaintance with the evidence furnished by the indicator. Permit me to call his attention to the fact that some people, probably as conversant as himself with the indicator, have been most woefully misled by a false interpretation of the indications furnished by the said indicator; *i. e.*, leaky exhaust valves of a steam cylinder have before now been taken as an indication of excessive cylinder condensation by men who were above the charge of ignorance.

"Mr. Lozano's position in stating that the indicator shows no re-evaporation of ammonia gas places him in a position subject to the charge of making the same error. While not trying to detract from the high position held by Mr. Lozano in the field of mechanical refrigeration, I must be allowed to differ from his assertion that the indicator would show any such result. In the first place, gas occluded in the oil is in a far different condition than gas compressed to the discharge pressure of the machine; compressed gas will undoubtedly re-expand as the piston starts on its return journey to the other end of the cylinder, and leave a record of its action on the card. But occluded gas, on the other hand, will not begin to re-expand until the pressure has been reduced to an appreciable extent—an operation that involves the element

of time, a period of time so short, in a card like the following, that no impression will be apparent, viz., time of piston travel from x to v. (In the sketch the line x v is intended to represent the time intervening between the beginning of the return stroke and the opening of the suction valve.) This time in a compressor running sixty revolutions per minute will be equivalent to the time occupied by the crank in traveling through not more than 10° of a great circle—one-thirty-sixth of one second. The amount of gas released in this time is so small that no one who is at all acquainted with the subject, and who has looked at it in this light, would for an instant expect to find any trace of its existence on any indicator card, no matter how sensitive or expensive an instrument was employed, either with or without a microscope. Mr. Lozano certainly deserves credit for the admission that 'internal cooling of an ammonia compressor by oil is little, if any, better than external cooling by water jacket.' It would certainly be very interesting to some of us 'plug-ugly' engineers if some one who is fully competent to show up the results of a test of some oil-injection machines, pumping wet gas, that I have seen in operation during



the past year, would publish the results of such a test and show up the proportions of the various sources of supply of gas as shown by the indicator; say the relative amounts supplied by the following: *x v w y*, representing the re-expansion of gas from oil, *y w a b*, representing the amount admitted by the suction valves, and *a b c d*, representing the gas generated from the liquid carried into the compressor by the incoming gas. I feel safe in asserting that the results would probably be a surprise to the builders of the machines subjected to such a test, yet the machines were run in accordance with the directions of the erecting engineers as understood by the engineers in charge.

"Undoubtedly the greatest loss occurs during the time that the suction valves are admitting the charge to the compressing cylinder. All or nearly all of the liquid entering the cylinder is evaporated upon coming in contact with the warmer walls of the cylinder and surface of the piston; each volume of gas so generated, while performing the duty of cooling the compressor, excludes a like volume of gas that has done useful work in the evaporating side of the machine, thereby reducing the capacity of the machine as a whole, but not reducing the power required to drive the compressor in anything like a corresponding amount. The action of an oil-injection machine is very similar, with the single excep-

tion that a cubic inch of gas expanding from the oil does not absorb as much heat as the same amount of gas expanding from liquid ammonia, and therefore does less in the way of cooling the cylinder than the wet gas alone, although as much power is consumed to compress the gas in one case as the other."

CHAPTER IX.

SUGGESTIONS TO ENGINEERS.

I HAVE thus far not given any rule or formula for figuring the capacity of an ammonia compressor for ice making or for refrigeration on the basis of compressor displacement; and it has been suggested that such a rule would be appreciated.

For upwards of fifteen years past I have been in the habit of applying for convenience a rule which I am inclined to believe my friend, Prof. Siebel, will immediately classify under his index heading (to "Compend of Mechanical Refrigeration") of "Rules of Thumb"; but as the rule was the result of careful observation, and as it has stood the test of practice for many years, I do not hesitate to assert that it is a safe one. I give it in the language of an old memorandum, just as noted down years ago, viz.:

"Four cubic feet, or 7,000 cubic inches, of compressor displacement per minute equals one ton of refrigerating capacity. Seven cubic feet, or 12,000 cubic inches, of compressor displacement per minute equals one ton of ice making capacity."

I recognize that there is a discrepancy between the values of the figures in the rules above given. It is small, however; and after checking up the rule by the results of many tests and the actual daily performance of many compressors working under varied conditions of pressures and temperatures, for both ice making and refrigerating purposes, I feel warranted in recommending the rule as convenient and safe in practice.

It may not be amiss to sandwich a few suggestions to the engineer in this article, while treating on the compressor subject. Every engineer should keep himself posted as to the exact clearance existing between the piston and the head or heads of his compressor. A convenient way to secure this information is to remove one of the valves from the compressor head and insert a piece of wire solder or soft sheet lead through the valve chamber into the cylinder and pinch the machine over until the piston of the compressor will squeeze the lead against the head. When the lead is withdrawn the exact clearance will be at once apparent, and can be measured on the lead. This method is quick and convenient, and is a help in equalizing the clearances at both ends of a double-acting compressor. Try it on your compressor, and regulate your clearances accordingly.

If you are troubled with your valves sticking in the compressor heads, it is generally

due to the lead gaskets pressing out against the heads and against the valve cages. A slight taper of the gasket seats on the cages, tapering the cage inward, will prevent this trouble, if the gaskets are properly trimmed before being put into the pump head.

If your valves pound, it is due to too much lift. Put a stiffer spring or a cushion spring that will prevent the valve opening beyond the area of the openings in your valve cages. If you get full area opening of the valve, you cannot better it by having an excess lift of the valve. Enough is as good as a feast.

A leaky suction valve is many times the cause of considerable loss of capacity. It can be located by heat on the suction connection, or by the hiss that is ever present when a valve leaks. If the machine is making too much noise to allow you to hear the hiss, take a wire in your teeth and touch the end of the wire to the valve cap, and you will have no trouble in locating the leak. A leaky discharge valve is as bad as a leaky suction valve, and it is harder to locate. The gas leaks down into the cylinder, and does not show itself by additional heat, nor is the hiss so easily discovered. You should have a duplicate set of valves on hand always in prime condition, and at convenient intervals change the valves in the compressor so that you may examine the valves at your leisure, and keep them in perfect condition.

Never allow a shoulder to remain in your cylinders. If the piston rings have formed a shoulder in the cylinder at the end of the stroke, file or scrape it out as soon as you discover it, and ascertain that your piston rod stands central in the guides at all points of the stroke. If the crosshead is gibbed out of line, it will throw the rod over, and is liable to make a shoulder in the cylinder through the piston's rocking. Remove the shoulder or you will have trouble later on, when you attempt to adjust your piston.

A very little lubrication will keep an ammonia compressor in good condition. Two or three drops of West Virginia oil per minute will answer for the largest vertical compressor, provided the cylinder is well oiled before starting up. Horizontal compressors, with oil circulation in the stuffing box, will take in sufficient oil to keep them well lubricated on the lower side of the cylinder through the stuffing box. The writer never saw a horizontal compressor opened up that was not dry on the upper side of the cylinder. You'd have to fill the cylinder with oil to wet a horizontal compressor on the upper side. Too much oil causes trouble in the apparatus outside of the compressor. Use enough for lubrication, but use it with discretion.

Use as little black oil as possible when testing your apparatus with air pressure. If you can get glycerine handy, don't use any

black oil under such circumstances. I have never known of an oil explosion in a plant that was charged with ammonia, but I have vivid recollections of several oil explosions when testing with air pressure; and as a choice between an oil explosion and a gunpowder explosion, I would prefer to be present at the latter. The natural mineral oils under high pressure and high temperature will produce a gas that is wonderfully quick and powerful. Its flash test is usually low, and when it flashes there is no time left for a man to dodge.

Never pump a vacuum on an ammonia system unless it is absolutely necessary to repair a leak, or for similar purposes. "Nature abhors a vacuum," and the tendency of a vacuum is to suck air into your plant. Keep the air out; it will not do you any good in your ammonia system. Pump to zero on the gauge, no lower.

The presence of weak liquor in the bottom of a single-acting compressor below the piston is sometimes the source of a great deal of annoyance. The packing in the stuffing boxes cannot be kept in good condition if the compressor contains weak liquor; the liquor will run down the rods and stink, and if the engineer tightens up the stuffing boxes, the rods will heat. The same conditions occur with double-acting vertical compressors at times; and when such conditions exist, there is

trouble for the engineer. He must watch the rods closely or run a chance of having the rods cut badly. The best way to get rid of the weak liquor is to take the packing out of the stuffing boxes and drain the compressors. If the lower heads are dished and a boss, or ridge, exists next to the rod to prevent scale or dirt from lodging against the rod, then the only sure way of getting rid of the weak liquor is to take out the piston and sponge the compressor out dry. Weak liquor will seek the lowest point and will lie in the bottom of the compressor, below the oil, if there is oil in the compressor, and it is one of the meanest evils an engineer has to contend with.

If you are operating a double-acting compressor, to do good work you must freeze over to the compressor. Without the compressor frosted, the cylinder and piston will get too hot to accomplish good results, and you are liable to burn out the packing. The wet compression seems to be the only practical way of operating a double-acting compressor. If you are operating a single-acting compressor *do not* freeze over to the compressor. You will do better work with dry compression in a single-acting compressor than it is possible to do with any other system of operating an ammonia compressor. If you freeze over to a single-acting compressor, you will freeze up the packing in your stuffing boxes, and you must in consequence watch the rods closely

or you will have trouble. When operating with wet compression, your discharge pipe should never be extremely hot, while, if you are operating with dry compression, your discharge pipe will get hot enough to burn your hand if held upon the pipe. There is more heat in the gas with dry compression than there is with wet compression, which fully explains the difference in temperature of the discharge pipe under the two conditions of operation.

If your cylinders are cut, you will necessarily have leakage of gas by the pistons, which means loss of capacity. Some manufacturers furnish a set of soft metal rings to be used when the cylinders are cut or scored. The soft metal will wear into the grooves and prevent, to a certain extent, the leakage of gas by the pistons. The soft rings, however, should not be used excepting when the cylinder is cut, as the soft metal will retain any dirt or scale that comes in contact with it, and the result will be the wearing or cutting of the cylinder a great deal more than with the ordinary cast iron rings. The best plan is to rebore a cut cylinder and fit it with a new set of cast iron rings as soon as possible after the cut is discovered.

Test the oil you are using in your compressors by subjecting a sample to low temperature. Part of a bottle of the oil suspended in the brine in your tank—or, better yet, surrounded by ice and salt—will demonstrate

whether it will stand a low temperature or not. If the oil gets thick and gummy and a separation occurs, leaving a thin, transparent, watery liquid, in which the body of the oil settles down and which gives off an odor like benzine, reject the oil, as it will produce permanent gases in your plant and give you lots of annoyance. If the oil merely thickens a little without any separation, and it will thin down again when subjected to higher temperature without giving off a bad odor, it can be used with safety. Some engineers test samples of oil with ammonia gas to determine if the oil contains animal matter. Animal fats will saponify when subjected to alkali tests. You do not need any animal matter in your ammonia system.

CHAPTER X.

OPERATING INSTRUCTIONS.

SOME builders of refrigerating machinery make it a practice to send out with each machine a printed pamphlet of instructions to engineers for the management, or operation, of that particular machine. Those who do not do so assume that the engineer placed in charge is familiar with their type of machinery, and needs no special instruction for the handling of a special make of machine. This, of course, as a rule is true; but as first-class refrigerating engineers are not as plenty as blackbirds in autumn, it occasionally happens that a man competent to handle a steam engine, but unfamiliar with refrigerating machinery, is placed in charge of a plant. Such a man may fairly ask the maker for his "book of instructions," and be disappointed, without doubt, when he gets the reply, "We regret to say that we have no book of instructions."

A builder of compression machines having received just such a request, and being compelled to make the above reply, "compromised" by sending his client the following, which I am permitted to reproduce as of

interest to engineers of "Consolidated" and "Boyle" machines especially, although the principles are applicable to other types of compression machines.

In the first place, the writer of these instructions advises the reader to obtain and read all books and periodicals devoted to the ice making and refrigerating industries, which will give the latest and best information on the subject. The questions are then taken up and disposed of as follows:

"*Question.*—Do you shut off the ammonia at the receiver when you shut off the engine?

"*Answer.*—When shutting down a machine, close the liquid valve on the brine tank, and run the machine slowly until you pump the pressure down to 0° on the gauge. Do not pump a vacuum, as a vacuum is liable to suck air into the pumps through the stuffing boxes. When the gauge shows 0°, there is actually atmospheric pressure of ammonia in the coils (14.7 pounds pressure), or, in other words, the pressure outside and inside is equalized.

"*Question.*—Do you start the engine before opening valve at receiver?

"*Answer.*—Yes; start the engine slowly, and before the pumps show a vacuum on coils, open the regulating valve slightly, and then speed up the engine until it works automatically; then open the regulating valve until the pressure on the low pressure gauge shows, say fifteen pounds above 0°, and keep the

pressure at that point; or, if you want to run the machine to full capacity, regulate the pressure until the frost shows on the suction pipe just above the brine tank. (Do not carry frost over to the pumps, or the stuffing boxes will freeze up and leak, and make you more or less trouble.)

"*Question.*—In case of leaking in the pipe, what do you do with the ammonia?

"*Answer.*—The connections are made to the pumps in such a manner that the large valves on the suction and discharge pipes may be closed and the small equalizing valves opened, and by running the machine slowly you can pump out the discharge pipe system into the suction pipe system and lock the ammonia up in the brine tank coils. In case of leakage on the suction side, it is only necessary to shut off the liquid valve and pump all of the ammonia up into the condenser and lock it in there by shutting the valve on the top of condenser.

"*Question.*—When the ammonia line is coated with oil, what steps do you take to get it out?

"*Answer.*—If the system gets clogged with oil, there is only one way to clean it out, *i. e.*, take all of the coils out of the tanks; blow them out with dry steam, and afterward with air to dry out any condensation that may deposit in the coils. Clean out the connections in the same way, and then connect up the coil

again. A solution of soda ash and water will also cut the oil out of the coils, but after using the soda ash it is necessary to blow out with steam and air to get rid of the soda ash. A good, careful engineer will never let his system get clogged with oil, as there are provisions made for purging oil from the oil interceptor, and if this is frequently purged there will be very little oil carried over into the coils. A liquid trap on the brine tank will positively prevent oil from entering the evaporating coils.

"*Question.*—How do you take ammonia from the drums of a 'Consolidated' machine? Do you make a connection to the pumps and pump through the line to the receiver?

"*Answer.*—There is a charging valve put on the liquid pipe, near the brine tank and between the regulating valve and evaporating coils, on every plant ever built by the Consolidated company. Connect the drum to the charging valve with a short piece of pipe; shut the regulating valve and open the charging valve wide open; then open the valve on the drum slightly and run the machine slowly until you pump a vacuum on the drum; then shut the charging valve and disconnect the drum and regulate pressure by the regulating valve in the same manner as before you put the new ammonia in. *Never put any ammonia in a plant until you have tested the ammonia by drawing a sample out of the drum and seeing that it will all evaporate and leave no residue.*

"*Question.*—How much pressure is the most economical on the ammonia line per square inch?

"*Answer.*—This is a question that would require further information in relation to surrounding conditions in order to make an intelligent and full answer. Generally speaking, however, we can say that the low pressure should be about 10 per cent of the condensing or high pressure. When you have from 150 to 180 pounds condensing pressure, you can carry from fifteen to eighteen pounds evaporating pressure economically. Pressures, however, depend entirely upon the temperatures of the matter surrounding the coils. When the brine is warm, you can carry high evaporating pressure without frosting back to the pumps; when the brine is cold, you can only carry low evaporating pressure without frosting to the pumps. The frost line showing just outside of the brine tank on the suction pipe is the very best indicator you can go by; as you know that when the frost shows just above the coils, every foot of coil surface is doing good work; and if you carry the frost line any further you are doing work that you get no benefit out from. Do all of the freezing in the brine tank, and do not waste work by frosting the suction pipe over to the pumps and having dripping pipes in the engine room. The condensing pressure depends upon the temperature of water used on the condenser.

Cold water means low condensing pressure, warm water means high condensing pressure. The table of pressures and temperatures in the 'Compend of Mechanical Refrigeration' will give you full information on both high and low pressures.

"*Question.*—Or any other information which I should have?

"*Answer.*—Keep your machine and engine room clean. Keep tab on your temperatures of brine and condensing water and evaporating and condensing pressures. Don't let your machine knock and pound, and don't key up until you get hot boxes. Exercise the same judgment in operating a refrigerating machine that you would devote to a marine or a Corliss engine. Cool judgment, or 'horse sense,' as it is usually termed, will locate and correct any difficulty in a refrigerating plant in short order. Experience will teach you more than any living engineer can tell you, and a few weeks of experience around a refrigerating plant will enable you to take care of it all right, if you are a licensed marine engineer."

CHAPTER XI.

COMPRESSOR EQUIPMENT.

THE connections to the ammonia compressor should be made in such a manner as to enable the heads of the compressor to be removed without necessitating the breaking of the pipe connections. Every engineer knows that it is advisable to examine the interior of his cylinders frequently; and he appreciates that it is considerable work to disconnect pipe joints every time he wishes to get into the cylinders. Especially is this a fact when dealing with the larger sizes of compressors and consequent larger sizes of pipe connections. There is absolutely no reason why a suction or discharge pipe connection should be made to the compressor heads. If the cylinders are properly designed, they will be provided with suitable ports leading into the suction and discharge chests or chambers in the compressor heads, in such a manner that the heads may be removed without disturbing the pipe connections. The pipe connections should always be made permanent to the cylinders. Few engines are constructed nowa-days with the steam and exhaust con-

nections made to the cylinder heads; and there is no more reason for making ammonia pipe connections to cylinder heads than there is for making steam connections to cylinder heads. Convenience in getting at the working parts of ammonia compressors is fully as desirable as convenience in getting at the working parts of a steam engine, and it is as practical to provide for this convenience in one case as it is in the other; but judging from the examples of complicated connections to ammonia compressor heads, as illustrated by the average design at present in use, the fact has escaped the attention of the manufacturer in the majority of instances; and it would appear that the effort has been in the direction of complicating rather than affording convenience and facility to the engineer. One of these days the engineer will take it into his head to dictate some *practical ideas* to the manufacturers, and when he does there will be some decided improvements made in many of the present constructions and connections to ammonia compressors.

Every compressor should be provided with what is known as "equalizing, by-pass or pump-over connections," which may be described as a system of small pipe connections, provided with suitable valves, for the purpose of pumping out the main discharge connections and delivering the gas into the suction-main connections, enabling the engineer to

evacuate the entire high pressure side of his apparatus without reversing the compressor valves. These connections are also very convenient at times when it is necessary for one man to handle and start up a large machine, as the pressure can be equalized on both sides of the compressor piston—a feature that the engineer will appreciate when it becomes necessary to pinch his engine off the center and get over to the throttle before the gas in the compressor cylinders can work the engine back on the center again. I will venture the assertion that many an engineer will remember occasions that have added tallies to his cuss list when he reads these words. Put on equalizing connections and quit cussing.

Another desirable feature that is too frequently overlooked in connecting up ammonia work is the almost absolute necessity for stop valves—*reliable stop valves*—at both ends of every individual part of the apparatus. Show me a cheap plant and I will point you out a plant that is "shy" of stop valves; and a plant that is short of means to cut out any individual detail will sooner or later be a tribulation to its operator. In the early days the Texas man was in the habit of carrying a revolver, not because he was sure to need it, but if he ever did need it he "needed it bad." The stop valve on the ammonia apparatus is a parallel condition, only that you *are sure* to need the valve some day; and if it isn't there

you will pay for it three times over before you are out of the woods.

The usual engine room connections on a compression plant are: A set of ammonia gauges, high and low pressure; an oil intercepter or separator; and a liquid receiver. Some manufacturers also add a low pressure drip, or oil separator, on the suction side of the apparatus—a connection more in the line of ornamentation than of utility; for if the oil ever works over to the evaporating surfaces, you might just as well let it pass on to the compressor, if it would ever travel that far under low temperature, which I doubt. The gauges are details that should receive careful attention, for in a great measure you depend upon them to indicate to you the conditions that exist inside of your apparatus, and you regulate your plant in accordance with the vibrations of the gauges. You should, therefore, pin your faith only to such instruments as you *know* to be reliable. Of late there have been many ammonia gauges offered on the market, quite a number of which were not at all suitable for the purpose for which they were offered. There are several reliable gauges, made by old, standard manufacturers, and you will have to pay those manufacturers a good price for a good article. Pay it and get what you want. The low priced article is of no value to you for any purpose.

The oil intercepter, or separator, is in-

tended to arrest any oil that may be carried over with the gas from the compressor before it reaches the condenser. It should be made long—the longer the better. It should be placed as far away from the compressor as convenience will allow. It should be provided with means for readily reaching its interior for cleansing—a removable bottom, flange for instance; and it should be purged frequently and kept as free from oil as possible. A second trap, placed below the oil intercepter and connected to the intercepter with a small pipe, provided with a suitable shut-off valve (to be closed only when purging the small trap), will keep the intercepter clean and prevent the oil from dashing over to the condenser if the intercepter is smaller than it should be. Some manufacturers put a glass gauge on the oil intercepter to indicate how much oil the intercepter contains at any given time. If your engineer allows a sufficient accumulation of oil in the intercepter to permit of its being seen through a glass gauge, it is time you took him out behind the boiler and talked to him of the error of his ways. No thorough engineer would allow sufficient oil to remain in the intercepter to ever show in a glass gauge. He would blow the oil out of the intercepter before it could give him away. It is sometimes desirable to cool an intercepter, by means of flowing water over it, to assist in separating the oil from the gas. This is a good

plan where the condensing pressure is extremely high or where the intercepter is scant in dimensions. The overflow water from the condenser can be used for the purpose in places where fresh water supply is scant.

The liquid receiver is merely a receptacle to hold the liquid ammonia which flows from the condenser. It should be made twice as large as is absolutely necessary to contain the charge of ammonia requisite for operating the plant, as one of the most dangerous features in connection with an ammonia apparatus is a liquid receiver *full of liquid.* I have attended the repairing of several plants that had been overcharged with ammonia liquid, or, in other words, that had been charged with more liquid ammonia than the receivers were capable of storing, and which had been shut down and the valves on the receivers closed when the receivers were *full* of liquid. In each instance the results were similar, the receivers burst —the last one, a receiver made of 12-inch pipe, ten feet long, with heavy cast iron heads, shot through a 24-inch brick wall and made as clean a hole as could be made with a cannon ball. The receiver should never be more than two-thirds full of liquid, and it would be dangerous to close the valves on a receiver unless the engineer was absolutely positive that the receiver was *not* flooded with liquid. Fill a boiler full of water and start a fire under it, and the boiler will burst. Fill a liquid re-

ceiver full of cool liquid and shut the valves, and the heat of the atmosphere will expand the liquid until the rupture occurs, and the resulting explosion is something terrific. If the valve between the receiver and the condenser is never closed, this danger is averted, as the condenser affords room for the expansion of the liquid; but if the liquid receiver is sufficiently large to hold double a reasonable charge of ammonia for the plant, you can lock all the ammonia into the receiver with safety.

The purpose of the condenser is indicated by its name. It is to convert a gas into a liquid. In principle there is nothing new in the construction of ammonia gas condensers. They are all modifications or adaptations of ideas that have for years been used in other lines of mechanics outside of the refrigerating and ice making field; and it may be said that every known form of condenser has been experimented with in connection with one or the other of the refrigerating branches of trade. That no difference in principle is involved in the process of condensing ammonia gas from that of condensing other volatile gases is a recognized and admitted fact, the only difference being the physical one of pressure required for the condensation of various gases at a given temperature, the principle remaining unchanged, regardless of the physical difference existing in the nature of the gas. Temperature and consequent pressure are

the only factors recognized in the process of condensation, and these factors must be given due consideration in the construction of every condenser, for whatsoever purpose the condenser is to be used.

The present practice in the construction of ammonia condensers can almost be covered by the use of two well known terms, the "submerged condenser" and the "atmospheric condenser." The submerged condenser is a system of coils or tubes entirely submerged in a tank of water, while the atmospheric condenser may be described as a system of coils or tubes erected in an open space where the atmosphere can be brought in contact with the coils, and the water can be trickled or sprayed over the coils. Theory would say that where the average temperature of the water is cooler than the average temperature of the atmosphere, the submerged condenser would be the most advisable to use; and where the average temperature of the atmosphere is cooler than the average temperature of the water, the atmospheric condenser should be given the preference. There are, however, features outside of the matter of average temperature which should be given consideration in either case. In theory and in practice surrounding conditions must always be given consideration, and construction should be adapted to best conform to all surrounding conditions.

For a number of years the writer held to the opinion that the submerged condenser was the most desirable form of condenser to be used in connection with an ammonia apparatus, and he has yet to be convinced that the submerged condenser is the inferior of the atmospheric condenser, an equivalent surface being allowed in each instance. The submerged condenser is usually constructed in such a manner that the discharge gas from the compressor enters the condensing coils at their highest point and travels down the coils, while the condensing water is injected into the tank at the lowest point. In consequence, the hottest gas enters the tank at the point where the water is the warmest, and gradually traveling down the coils, meets with a countercurrent of cooler water until when liquefied the ammonia leaves the condenser at the temperature of the water entering the condenser —a more natural interchange than is possible to accomplish with the atmospheric condenser, for the reason that it is impossible to create reverse currents as between the gas and the water in an atmospheric condenser, the water refusing to accommodate us by climbing up the coils; and we are, in consequence, compelled to allow it to follow nature's dictates and flow down the coils in the same lines that the gas is following inside the coils. The liquid from the atmospheric condenser in consequence leaves the condenser at the same

temperature as the warmest water leaves it, and the interchange is not so natural or perfect as in the case of the submerged condenser.

It is claimed that the atmospheric condenser uses less water to accomplish a given result than the submerged condenser requires. I do not admit the truth of the claim, nor am I in a position to positively deny it. I have never seen a submerged condenser equipped with anything like the amount of surface that is allowed per ton of rated capacity for the atmospheric condenser; and until I am in a position to test the comparative merits of the two forms of condensers on a basis of equal footing for each, I will not condemn "the little fellow that does good work" (as one of my old engineer friends is pleased to describe his submerged condenser) to the advantage of the atmospheric condenser.

I am free to admit that with the atmospheric condenser there are features that are very convenient and desirable. It is easy to get at the coils to clean them, or to make repairs, or to stop leaky joints. The coils do not have to be handled in and out of a tank when it is necessary to blow them out and cleanse them internally. If the coils leak, it is discovered and the leaks can be promptly repaired. Mud and fungus do not collect on the coils and raise the condensing pressure, as is sometimes the case with the submerged

condenser. Evaporation of condensing water on the coils may act as an assistant to the condensing of the gas, and the atmosphere does undoubtedly help do the work; nevertheless, the fact cannot be disposed of that the construction of the atmospheric condenser cannot be made as near theoretically correct as can the submerged condenser. To make a submerged condenser with exposed surface as great as is usually allowed for an atmospheric condenser would cost nearly or quite double the cost of the atmospheric condenser; and right here is the secret of the favor that is shown to the atmospheric condenser by the trade in general at the present time. The purchaser wants all he can get for his money, and the contractor wants all the profit he can get out of his trade; and there you are.

In recommending pipe for condensing surfaces I have only to repeat what I have said in former chapters in these pages: the smaller the diameter of the pipe used, the more surface will be exposed proportionate to the contents of the pipe. That one statement tells the whole story of the value of small pipe for heat transmitting surface. Don't use large pipe for heat transmitting surfaces anywhere about your plant.

CHAPTER XII.

AMMONIA CONDENSERS.

TO give my readers some idea of ammonia condenser construction and the relative amount of pipe surface exposed per ton of refrigerating and ice making nominal capacity, I have prepared some tables * taking the same from a list of upwards of two hundred plants in actual operation, manufactured by several different machine companies; and in compiling the tables I have endeavored to give the *exact* dimensions of all parts of the condensers. The conclusion to be derived from an examination of the tables is, that there is a considerable variation of opinion existing as to the amount of surface which should be allowed per ton of duty to be performed. My own opinion, based upon my experience in the business, is that the largest condenser listed in the tables is none too large for economical results; in fact, I have never seen an ammonia condenser that was, in my opinion, of greater exposed surface than would seem desirable. I think I am warranted in recommending at least 200 feet of 1-inch pipe

* For tables see page 146.

per ton of ice making capacity, or 100 feet of 1-inch pipe per ton of refrigerating capacity, for submerged ammonia condensers, and 300 feet of 1-inch pipe per ton of ice making capacity, or 150 feet of 1-inch pipe per ton of refrigerating capacity, for atmospheric ammonia condensers; and I would base these recommendations upon the results obtained with condensers in actual operation on both refrigerating and ice making plants. For larger sizes of pipe, the relative exposed surfaces should be at least equivalent; and I would feel inclined to say even more, proportionately as the volume of the contents of the pipe increases relative to the surface exposed.

I have seen ammonia condensers made of nearly every size of pipe from 1-inch to 6-inch, and of forms and proportions, to say the least, very peculiar—some submerged condensers with straight, vertical pipes not exceeding six feet long, precluding any material travel of the gas; others, with but one continuous coil, the same size as the discharge from the compressor, winding to such length that liquid would form half way up the coil and necessitate draining the liquid down a tortuous path before it could be released from the condenser. A happy medium between these two faulty constructions has been recognized and adopted by a majority of the builders long since, but there are still to be found adherents to the cheap, spiral coil construction.

Upwards of fifteen years ago a Frenchman, with an eye to utilizing waste material that he had on hand, converted two old heaters into an ammonia condenser. He placed one shell above the other and connected the two together at both steam and water spaces; he discharged the ammonia gas in the space around the tubes in the upper shell, and fed the condensing water through the tubes, the water entering the lower shell first and then passing up through and overflowing from the upper shell. The tubes were expanded and beaded into the heads in the ordinary manner, and they leaked more or less (considerably more than less); but the old "Frank" managed to pull through one season and made some money with a plant that had been a "Jonah" to every one who had touched it before that. The old man was a mechanic from hat to shoes. He recognized the principle involved, and when he did not have money to get such a condenser as he wanted, he used what he had at hand and made it answer his purpose. He knew that a condenser was but a temperature exchanger, and he recognized in the two old heaters articles that had been constructed for that very purpose, and he made use of them. The profession lost an artist when the old man exchanged his ice tongs for a pair of wings, back in '81.

In the early '80's, D. L. Holden, an old thorough-bred in the business, a man that would

lose his meals and his sleep (to say nothing about his dollars) to demonstrate any idea he got into his wise old head in relation to ammonia or ice machinery, built a submerged condenser that came pretty near being the correct thing, according to the theory of the business. He made a straight shell, if I remember rightly, about thirty feet high, and put straight, vertical tubes into the shell, connecting the tubes at top and bottom with a chamber, the top chamber to receive the discharge gas from the compressor, and the bottom chamber to act as a liquid receiver. The water entered the shell at the bottom and gradually rose, meeting the current of warm gas as it rose, and creating an interchange of temperature between the gas and the water that was pretty near what it should be. That condenser was correct in principle, and it was only a question of experiment to demonstrate what length of shell would be the most desirable and practical (suitable surface being granted) to have reached perfection with submerged condensers. As fast as liquefaction occurred in that condenser, the liquid found its way down into the receiver without hindrance—it was a straight drop to the liquid receiver; the liquid would leave the condenser as cold as the water entering the shell; the water would overflow from the condenser very nearly as warm as the discharge gas from the compressor; and I believe the condenser would operate with less

condensing water, proportionate to its capacity, than any other submerged condenser ever made. Holden was an expert in many other branches of the art, outside of condensers; and many a man who is to-day considered an authority on ice machines has gained no little of his knowledge through the open and thorough explanations given him by D. L. Holden. There are modern machines on the market to-day that incorporate ideas born in the brain of Holden many years ago, and for which he received little or nothing from the parties who benefited by his ingenuity. I bow to Holden.

T. L. Rankin is another old timer whose handiwork can be detected in many of the devices at present in vogue in connection with ice making and refrigerating machinery and apparatus. "Tom" has had more experience to the square inch than any three ice machine men in the world, and he is a long way from quitting yet. He has made and unmade fortunes in the business, and I look for him to make another fortune before he lays down the test tube and picks up the harp. Rankin has made condensers and condensers and more of them. One that I shall always remember—and I guess he will, too—was a sort of a combination submerged-atmospheric condenser, which might be described as a coil within a coil, the outside coil being made of large pipe and the inside coil of smaller pipe, leaving a space between the internal and external coils. Into this space

the gas was discharged from the compressors; the inside coil carried the condensing water from the bottom upward, and the water overflowed and trickled down the exterior of the outside coil, so that the gas was exposed to both an internal and external condensing surface, besides being spread out to the utmost limit by reason of the narrow space between the two pipes. The ends of the inside pipes passed out through stuffing boxes in the bends of the external coil in such a manner that the inside coil could be taken out at any time and repaired, or renewed if necessary. The condenser was a good one; but like all good things, "it came high."

To undertake to review all of Rankin's work on the condensers, or, for that matter, on any parts of ice machines, compression or absorption, would mean the appropriation of the entire space of a volume much larger than this, and then the half of it could not be told. Tom Rankin is a genius, wedded to his profession, happy in his achievements and successes, and still happy if the game goes against him. "Tom" would take a contract for a skating rink in Hades or a toboggan slide into the crater of Vesuvius, and the chances are that he would come out smiling from either job. Lots of us owe a debt of gratitude to "Tom" Rankin, both for information and for fun and good fellowship. "Here's looking at you, old boy."

I have on file a list of a large number of 2-inch atmospheric condensers, running all the way from fifty-eight feet to 160 feet of 2-inch pipe to the ton of refrigerating capacity, built by several different concerns, both with and without draining connections for carrying off the liquid at different points of the coil. The majority of the coils are built similar to a "Baudelot cooler" in construction, usually made up with screwed return bends, soldered on the pipes; flat coils, about twenty pipes high and twenty feet long, more or less, as the occasion requires, or, rather, as the humor of the manufacturer seems to dictate; but one firm in my knowledge has adopted and stuck to a standard for 2-inch condenser surfaces for any length of time past. The wide deviation between fifty-eight feet and 160 feet of 2-inch pipe to the ton of refrigerating capacity illustrates fully the difference of opinion that exists in the trade in relation to the value of 2-inch pipe for heat transmitting surface in ammonia condensers. One or the other must of necessity be wrong. Being prejudiced myself against large pipe for heat transmitting surface in any part of a refrigerating plant, I shall leave it to the reader to make his own comparisons of relative surfaces as compared with the surfaces given in the tables published herewith, and let him draw his own conclusions from the comparisons.

Many manufacturers have lately adopted

"forecoolers," or "supplementary condensers," in connection with the condenser proper. This apparatus consists of a coil, or series of coils, through which the discharge gas passes before reaching the condenser proper, and by means of which the gas is cooled by utilizing the overflow water from the condenser proper. I should think the term "forecooler" more appropriate for this apparatus than the name "supplementary condenser," because it is not expected or desired to condense any gas in this part of the apparatus; it is merely intended to cool the gas before it is delivered to the condenser.

So far as the writer knows, the origin of this apparatus was evolved in the brain of a brewer, Mr. Ellis Wainwright, of St. Louis, Mo., who was, I believe, the first man to put this adjunct to a refrigerating plant in practical operation. He had a 25-ton plant in his original brewery, and instructed his engineer to put a coil of pipe between the compressors and the condenser "to utilize the overflow water to the fullest extent." Such is the language of a memorandum of a report made to the writer in 1881; and it would appear that to Mr. Wainwright belongs the credit of an invention that several parties have tried, unsuccessfully, to patent on different occasions since his apparatus was put in operation. The "forecooler" is a good thing. I have used it to advantage many times when condensing

surfaces were too limited or when the condensing water was short or very warm. It is an article that can be attached to any plant, and the cost is not very high, a factor that is generally taken into consideration. Several designs of atmospheric condensers have the "forecooler" connected below the condensing coils, in the same pan with the condenser, and the condensing water flows down over the "forecooler" before dropping into the pan. Others are arranged with a condenser pan deep enough to submerge the "forecooler" coils in the overflow water from the condenser; still others are made entirely separate from the condenser, and are designed to operate on either the atmospheric or the submerged principle, as may best suit the conditions and surroundings. Some are merely a coil of pipe the same size as the discharge pipe from the compressors, but the best are small pipe coils with area openings sufficient to balance the area of the discharge pipe. If you have high condensing pressure, insufficient condensing surface, scarcity of water, or warm water supply, you will find the "forecooler" a valuable aid to your apparatus.

Every condenser should be provided with means for purging air or permanent gases from the same. Any plant is liable to become contaminated with these troublesome and undesirable elements; and if the means of ridding the plant of the same is not provided it is

next to impossible to operate satisfactorily or successfully once the system becomes permeated with them. The purge valve should be placed at the highest point on the discharge line—as near the condenser as possible—and the purging should be done when the machine is not running. The water should be kept running over the condenser, and sufficient time allowed for the condenser to cool off before the purge valve should be opened at all. A pipe carried to a barrel of water from the purge valve or down into the water in the tank, in case the condenser is of the submerged type, will enable the engineer to purge the permanent gas from the plant without losing any material amount of ammonia gas. So long as permanent gases or air are being discharged from the condenser, large bubbles will rise to the surface of the water, with merely a bubbling noise or sound, but when ammonia gas begins to flow from the purge pipe into the water, there will result a snapping and crackling like the results of admitting hot steam into a cold pipe. It is not at all necessary to open the purge valve wide; a very little opening will allow the gases to escape, and the less the valve is opened the quicker it can be closed when the ammonia begins to escape. In this manner, by frequently repeating the operation, air or permanent gases can be purged from a condenser. It cannot, however, be done all at once, for the

reason that when a plant contains permanent gases, it is very liable to have the gases trapped in all parts of the apparatus, and it is necessary to gradually pump it all into the condenser and allow it to separate and seek the highest point (where it will finally lodge) before it can be got rid of.

I have seen a few plants with air chambers, placed at the highest point on the discharge line, to act as reservoirs for the collection of air or permanent gases while the machine is in operation. While such chambers do no harm on the plant, I am of the opinion that they do not collect much air when the plant is running. The discharge from the compressors is too strong to admit of a separation taking place while the machine is in motion; a circulation is bound to be going on throughout the discharge side of the plant, and if air or permanent gas collects in the chamber, it is liable to be driven out by the ammonia gas from the compressors. That the trap will come in play all right if the machine is shut down and the water allowed to run over the condenser, I am willing to admit; and that it might be connected so as to purge the chamber entirely without any possibility of losing ammonia from the condenser, is also a fact. It would only be necessary to provide a valve between the condenser and the chamber, to be closed when the chamber is to be purged, to make further loss of gas from the condenser than that possibly held in the chamber an impossibility.

I have seen quite a number of specifications for condensers in the past year or so requiring a valve on each end of each coil of the condenser, a requirement that I would consider more in the line of a convenience than an absolute necessity. Nearly all condensers are provided with a main valve at top and bottom in these days; and if the condenser is properly constructed the liquid will drain to the liquid receiver as fast as it is formed. The aim is to relieve the condenser of liquid as quickly as possible and utilize the space in the condenser for gas, not to store and hold liquid in the condenser. The condenser can be shut off from the liquid receiver and from the compressors and discharge line in case of accident; or when slight leaks occur the condenser can be pumped out and repairs made without a great deal of trouble or loss of time. The valve on each end of each coil would merely cut out the individual coil and lessen the condenser capacity to that extent, while to make necessary repairs on the individual coil would, in three times out of five, require that the entire condenser should be pumped out before the coil could be safely disconnected and the repair made.

I recommend valves wherever necessary; but I think a majority of the practical engineers would be perfectly satisfied to have their condensers constructed without a valve on each end of every coil, unless the coils were

constructed on the "Baudelot cooler" principle, in which case I admit the necessity of a valve on each end of each coil.

DIMENSIONS OF ATMOSPHERIC CONDENSERS.

Ice Making Capacity. In Tons.	Refrigerating Capacity. In Tons.	Condenser Pans. Length of Pan. Feet.	Width of Pan. Feet.	Depth of Pan. Inches.	Thickness of Iron. Inches.	Number of Pipes High.	Number of Pipes Wide.	Size of Pipe. Inches.	Length of Coil over Bends. Feet.	Total Feet of Pipe in Condenser.	Feet of Pipe per Ton, Ice Making Capacity.	Feet of Pipe per Ton, Refrigerating Capacity.
12½	25	21	10⅔	8	3-16	40	5	1	17	3,680	294.4	147.2
20	35	24½	10⅔	8	3-16	40	5	1	21	4,440	222.	126.8
30	50	24½	14	8	3-16	50	7	1	21	7,750	258.3	155.
40	75	24½	14	8	3-16	50	7	1¼	21	7,750	193.75	103 33
50	100	24½	14	8	3-16	90	7	1	21	13,950	279.	139.5
60	125	24½	14	12	3-16	80	7	1¼	21	12,400	206.6	99.2
80	150	27½	17	12	3-16	80	7	1¼	24	14,080	176.	93.86
		Aver	age	for	1-	in.	Pip	e	per	ton,	263.42	142.12
		Aver	age	for	1¼-	in.	Pip	e	per	ton,	192.12	98.79

DIMENSIONS OF SUBMERGED CONDENSERS.

Ice Making Capacity. In Tons.	Refrigerating Capacity. In Tons.	Tanks. Length. Feet.	Width. Feet.	Depth. Feet.	Thickness of Iron. Inches.	Number of Coils.	Pipes High.	Feet Long.	Size of Pipe. Inches.	Total Feet of Pipe in Condenser.	Feet of Pipe per Ton. Ice Making Capacity.	Feet of Pipe per Ton. Refrigerating Capacity.
5	10	10	3⅓	6½	3-16	9	12	7½	1	855	171.	85.5
10	20	10	7½	6½	3-16	20	12	7½	1	1,900	190.	95.
12½	25	10	7½	6½	3-16	22	12	7½	1	2,090	167.	83.6
15	30	10	8⅓	6½	3-16	25	12	7½	1	2,375	151.6	79 16
20	35	10	10	6½	3-16	27	12	7½	1	2,565	128.25	73.28
30	50	10	10	12½	¼	27	24	7½	1	5,130	171.	102.6
40	75	14	10	13½	⅜	27	24	11½	1	7,695	191.1	102.6
60	110	14	13	13½	⅜	35	24	11½	1	9,975	166.25	90.68
								Av.	er	age,	167.	89.

CHAPTER XIII.

MAKING ICE.

THERE is considerable difference between capacities of machines for ice making and for refrigerating. To illustrate this difference we take a given quantity of brine to be cooled from say 80° F. to say 14° F. Not taking into consideration the specific heat of the brine as compared with fresh water (the average brine having only about 79 per cent of the specific heat of fresh water), sixty-six British thermal units of heat would have to be removed from each pound of brine in the process of cooling the same from 80° to 14° F. Taking fresh water at 80° F. and freezing the same, and then cooling the ice down to 14° F., covers a much greater amount of work than is represented by the process of cooling brine through the same range of temperatures. First, the water must be cooled from 80° to 32° F., requiring the removal of forty-eight British thermal units of heat from each pound of water; next, the latent heat of the water must be removed before the water will congeal into ice. This process requires the removal of 142.6 British thermal units of heat

from each pound of water in order to change the water at 32° F. into ice at 32° F.; and to cool the ice from 32° to 14° F. requires the additional work of removing eighteen British thermal units of heat (not taking into consideration the specific heat of ice, which is only 50 per cent of the specific heat of water). The total number of units of heat to be removed in the case of the ice making is, therefore, 208.6 thermal units as against sixty-six thermal units for refrigerating brine through the same range of temperatures; and the secret of the difference is found in the fact that in the refrigerating process the latent heat is not removed from the brine, while in the ice making process the latent heat is removed from the water in the freezing.

This illustration would seem to indicate that a plant for ice making purposes should be a little over three times the size of a plant to be used for refrigerating purposes; but refrigerating capacity of a plant is figured on a basis of the refrigeration performed by the melting of a given quantity of ice at 32° to water at 32° F.: in reality the latent heat basis. The refrigerating capacity of a plant is, therefore, merely nominal—an accepted term applied in much the same manner as the term "horse power" is applied when rating the capacity of a steam engine, while ice making capacity of a plant is an actuality, capable of being demonstrated practically on a scale. We can weigh

the ice we are making, but we have to figure the refrigerating capacity of a plant. Ordinarily it is assumed that a given size of machine will produce ice to the extent of 50 per cent of its nominal refrigerating capacity; but as the assumption is like comparing facts with figures, the comparison is actually one of very little value in establishing a ratio of the values of ice making and refrigerating capacities. One is actual, the other is a nominal capacity.

In treating the subject of ice making, I shall endeavor to give my readers the results of actual experiences, and will not attempt to advance any theories, either of my own or of others, to mislead them into experiments in ice making. Thousands of dollars—fortunes—have been spent in experiments in the ice making field, and almost every year some visionary inventive genius will add to the pile by erecting a monument to his folly when, by merely looking around and taking advantage of the lessons to be derived from the examples of numerous successful ice making plants in operation in all parts of the country, he could have invested his time and money to greater advantage, and at the same time put in a great many more hours sleeping well that were wasted in trying to accomplish something that no one else had tried and that would not be worth anything in particular if accomplished. Some other fellow invariably follows over the trail of the experimenters and

profits by the errors the experimenters have made, and the owner of the plant usually has both the experimenter and his follower to vent his spleen upon; and the worst part of the whole business is that the owner generally is the one who has to stand the losses consequent upon the experiments. My advice to the buyer is to find a plant that is giving satisfaction and paying dividends to the stockholders, and then follow that construction in the construction of the plant he purchases. Don't buy an experiment. It will prove as expensive as a lawsuit, and will probably get you involved in a lawsuit before the contract is completed.

My own first experiences in ice making were with plate plants, under the tuition of one of the soundest authorities in the country, the late David Boyle, a man who was at least the equal, if not the superior, of any ice making engineer in the world. To David Boyle rightfully belongs the credit of having produced the most successful ice making plants extant in the early days of the art, and at the present time there is hardly a plant in this country, constructed on either the plate or can systems, that has not incorporated in its construction some detail evolved in the brain of that tireless and studious Scotchman. I have known David Boyle to travel many miles just to examine an ice making plant built by some competitor, and not a detail of the construction of the plant would escape his watch-

ful eye. He would spend hours and hours in the public libraries, poring over any literature he could find bearing upon ice making and refrigerating and the transmission of heat. Mr. Boyle thoroughly understood the principles involved in the manufacture of ice, and was undoubtedly the first man to adapt a machine and apparatus to the practical demonstration of those principles. His first plants were on the plate system, with two separate freezing tanks, each containing hollow iron plates, through which the refrigerating agent was circulated. The cooling of the refrigerant was accomplished in a third tank containing the evaporating coils, and the circulation of the refrigerant was by means of a pump driven by power taken from the machine. Later he abandoned the separate evaporator tank and placed the evaporator coils within the hollow plates in the freezing tanks, thereby economizing the space required for the plant and also decreasing the loss due to radiation, which was a considerable item with the separate evaporating tank. So far as results are concerned, David Boyle's early plate plants would compare very favorably with the larger modern plate plants, and I can say that I have not seen anything in the line of a modern plate plant that is not, to a greater or less extent, merely a copy or modification of David Boyle's early construction; certainly no new idea has been evolved in any of the later plate or block

systems, unless the methods of removing the ice from the plates or coils are considered; and I am free to confess that I can recognize no economy in the present accepted methods of removing the ice from the plates or coils over the methods employed by David Boyle for the purpose; in fact, I am of the opinion that Boyle's old methods would prove more economical than the present steam-thawing methods.

Mr. Boyle appreciated the fact that the plate system was not the most practical and economical system for making ice; and upwards of twenty years ago he set about to devise means for doing away with the plates and constructing a plant that would be continuous, regular and steady in its operation, and one in which the conditions of operation would not materially alter during the process of harvesting the product—a condition that exists in every plate or block plant. He had experimented more or less on the can system, and had visited a number of plants constructed on this system, all of which he found to be making white or cloudy ice, very few using a distilling apparatus, and such as did attempt to distill the water had apparatus so crude that the distillation and purification were but partially accomplished, and the ice in consequence did not compare in quality with the product of his own plate plants. Mr. Boyle said that he would make a distilling apparatus that would

produce pure distilled water for ice making purposes, and with the persistency for which the race from which he sprang is noted, he kept at the problem until his efforts were crowned with success. He made the first really pure distilled water ice, and now the procession is following the trail he blazed through the wilderness in the business. "Dave" has gone, but his name and his works will live as long as ice making is a recognized industry.

From experience, observation and general repute, I am prejudiced against the plate system of ice making, and yet there are conditions of surroundings under which I would be tempted to construct and operate a plate plant; as, for instance, where fuel is high priced and an abundant water power is available, I would be inclined to put up with the annoyances and inconveniences of the plate system, unless I could get a price for my product in keeping with the additional expense that would be incurred if the ice was to be manufactured from distilled water. But where cheap fuel is available, my preference will always be with the can system. My reasons for this preference are that from experience, observation and inquiry I have satisfied myself that the plate system requires much larger compressors and surfaces for evaporating and condensing. I find that plate plants are constructed with an allowance of about 20,000 cubic inches of compressor displacement per minute per ton of

ice making capacity, while practice has demonstrated that an allowance of from 12,000 to 14,000 cubic inches of compressor displacement per minute per ton of ice making capacity is amply sufficient for can system plants. The allowance for condensing and evaporating surfaces with the two systems will hold good to very nearly the same ratios as are expressed by the ratios of the compressor displacements. It is apparent, therefore, that the plate system is necessarily a more expensive system to construct than is the can system. The plate system is obviously an intermittent system, while the can system is a regular, continuous system, operating under the same conditions of temperature and consequent pressure at all times, a factor worthy of the greatest consideration when it is appreciated that in order to make ice regularly and of uniform quality the regularity of the temperature and pressure is an absolute essential to the results. The can system delivers the product shaped in the most convenient form for the trade, while the product of the plate system has to be cut to the shape required. The can system freezes the ice from two sides toward the center, while the plate system freezes the ice from one surface only. It is obvious that the can system will freeze much faster in proportion to the thickness of the product than is possible with the plate system. With a good distilling apparatus a

can system will produce absolutely chemically pure ice, while the plate system will freeze into the ice any impurities held in solution in the water which are not precipitated by the action of low temperature. Outside the one factor of avoiding the necessity of distilling the water, I contend that there is not a single advantage to be derived from the use of a plate system over a can system. The plate system will make good commercial ice from natural water, free from the feather center that is always present in can ice; but all things being considered, I am of the opinion that the plate ice will cost fully as much per ton as the can ice to manufacture, and I know the can system is a much simpler and easier plant to handle and operate successfully than is the plate system.

From memoranda in my possession I have compiled a table of standard ice making tanks constructed on the can system, each and every one of which "has been tried and not found wanting"; and I do not, therefore, hesitate in recommending the tanks as reliable, when operated in connection with any first-class compressors of suitable displacement for the tonnage represented in the table. There is not a single tank listed that has not produced more than its nominal rated capacity of first quality can ice. I have been asked many times if I do not consider longer hours' allowance for freezing to be advantageous with a can plant.

ICE MAKING TANKS.

Tons Ice Making Capacity.	No. of Tanks.	Sizes of Tanks.				Number of Coils.	Size of Pipe, Inches.	No. of Pipes High.	Length of Coils.	Total Feet of Pipe in Tank.	Feet of Pipe per Ton Ice Making Capacity.	Number of Ice Molds in Tank.	Size of Molds in Inches.	Net Weight of Ice from Each Mold.	Number of Molds per Ton Ice Making Capacity.	Number Hours for Freezing Each Mold.	Remarks.
		Length of Tank, Feet and Inches.	Width of Tank, Feet and Inches.	Depth of Tank in Inches.	Thickness of Plates, Inches.												
2	1	17—0	6—2	33	3-16	7	1	6	15—4	644	322	60	8×15×33	100 lbs.	30	36	
3	1	17—0	9—0	33	3-16	10	1	6	15—0	900	300	90	8×15×33	100 "	30	36	
5	1	17—0	14—9	33	3-16	16	1	6	15—0	1,440	288	150	8×15×33	100 "	30	36	
10	1	29—0	19—0	33	¼	25	1	8	17—0	3,400	340	192	11×22×33	200 "	20.4	48	
												24	11×11×33	100 "			
12½	1	37—6	19—0	33	¼	33	1	8	17—0	4,488	329	256	11×22×33	200 "	21.36	52.2	Special.
												32	11×11×22	100 "			
15	1	43—0	19—0	33	¼	37	1	8	17—0	5,032	335	288	11×22×33	200 "	20.4	48	
												36	11×11×33	100 "			
20	2	29—0	19—0	33	¼	25	1	8	17—0	3,400	340	192	11×22×33	200 "	20.4	48	*
												24	11×11×33	100 "			
30	2	43—0	19—0	33	¼	37	1	8	17—0	5,032	335	288	11×22×33	200 "	20.4	48	*
												36	11×11×33	100 "			
30	1	43—0	30—0	48	¼	35	1¼	8	28—0	7,840	261	480	11×22×45	300 "	16	57.6	
60	2	56—0	20—0	48	¼	49	1¼	10	18—0	8,820	294	432	11×22×45	300 "	14.4	51.8	
60	2	43—0	30—0	48	¼	35	1¼	8	28—0	7,840	261	480	11×22×45	300 "	16	57.6	*

Average of 1-inch pipe per ton, 327 feet. Average of 1¼-inch pipe per ton, 272 feet.

*Twenty-ton tanks are duplicate 10-ton tanks
Thirty-ton " " " 15 " "
Sixty-ton " " " 30 " "
} Dimensions of one tank only are given in each instance.

My reply has invariably been, yes. I know that the slower the ice is frozen the better its appearance will be. I further recognize the advantage of large freezing tanks, when it is desired to force a plant beyond its rated capacity to meet the demand for ice in very hot weather, a condition that frequently tempts the proprietor of the ice factory to "rob his tanks" and regret it later. The tanks I have listed are, however, amply large enough to produce their rated capacity of first-class ice with a brine temperature of 14° F., and that is about the temperature the average engineer endeavors to carry, and succeeds in carrying, if he compels the ice pullers to take out the ice regularly, and does not allow his "boss" to "rob the tanks."

I appreciate how hard it is for the proprietor of an ice factory to see a crowd at the ice house door with their change in their hands, begging for ice when there is no ice in the ice house and only one block at a time coming out of the tank at stated intervals; but if I was chief engineer of an ice factory and any man attempted to "rob my tanks" I'd quit my job or chase that man out of the tank room with an ice ax before I would allow him to take a single can out of the tank before its regular time.

CHAPTER XIV.

HINTS ON ICE MAKING.

THE same opportunity exists for covering up defective work in the freezing tank that is often discovered in refrigerating brine tanks. The evaporating surfaces are submerged in a similar manner in both cases; consequently defects are not easily discovered, and when discovered are necessarily difficult to reach and repair, frequently requiring the emptying of the tank before the defect can be uncovered and attended to. Too much care cannot, therefore, be devoted to the construction of the freezing surfaces and their connections.

Freezing coils should invariably be made of the best quality of extra strong pipe, welded and bent into continuous lengths whenever practicable, avoiding screwed return bends and the consequent liability of leaks from numerous joints. The bent coil reduces the number of joints to the minimum, and the fewer the joints in a submerged system the less liability for leaks. Common pipe coils should *never* be used in connection with any part of an ammonia system. They are not

reliable, and certainly are not economical, if the factor of safety is given due consideration. I know of cases where common pipe coils have cost more than their value in loss of ammonia alone, and then had to be replaced with extra strong pipe coils before the plant could be operated satisfactorily. The best steel ammonia fittings should be used for connecting the coils to the manifolds; the cheap, light fittings are as unreliable as the common pipe coils, and should always be avoided when making joints which are to be submerged.

Careful attention should be given to the matter of area openings to manifolds in freezing tanks, as it is frequently necessary to put a greater number of coils in a freezing tank than the area of the suction pipe to the compressor will handle without choking the openings from the coils to the manifolds until the areas of the openings and the area of the suction pipe are equalized. I have found more trouble arising from the source of unequal areas, or, rather, excessive areas, of manifold openings, than from any other source in connection with freezing tanks, and I have invariably found that when the areas of the openings were reduced until they equalized with the area of the suction pipe, the trouble was at once obviated.

To illustrate the point under discussion, let us suppose that we have a 3-inch extra-strong suction pipe, and the coils in our freez-

ing tank are to be made of 1-inch extra-strong pipe. The area of the 3-inch suction pipe is 6.56 square inches; the area of one 1-inch pipe is .71 square inches. A division will show that nine 1-inch coils would supply all of the gas that the suction pipe is capable of handling. For the sake of the argument, let us suppose that we find it necessary to use eighteen freezing coils in our tank; if we leave full area openings, we know that one-half of the coils will supply all of the gas that the suction pipe can handle, and in consequence one-half of the coils must necessarily be devoid of circulation, and the gas will back and fill in one-half of the coils as against the other half of the coils, causing irregular and unsatisfactory freezing; but if we reduce the area of the openings from the coils to the suction pipe to one-half the full area of the coil pipes, we will have equalized the areas of the openings to the area of the suction pipe, and the result will be that *every coil in the tank will do its proportionate amount of work in furnishing the suction pipe with gas, and there will be an active circulation of gas in every coil in the tank* without any part of the coils working against the other coils.

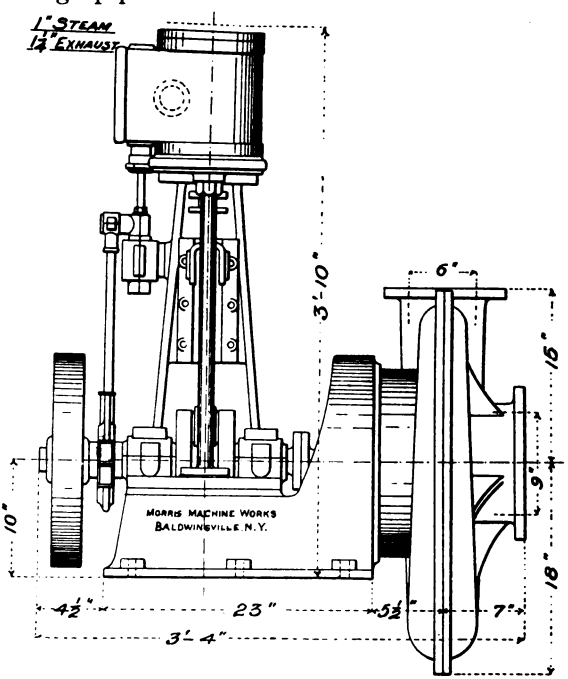
Equal circulation throughout the evaporating and expansion surfaces is one of the most important factors of successful ice making. It is the key note to steady, regular work; and where the circulation is not steady and regu-

lar, we must not expect to get steady, regular results. I have found the top feed and bottom expansion manifold system to be one of the very best arrangements to secure equivalent area openings without undue choking of the coil areas. This system is merely traveling the gas down one coil and up the next coil, or doubling the travel of the gas and thereby reducing the openings to the suction to one-half the number of the coils in the tank. This system does not retard the circulation of the gas in the least; it merely doubles the travel of the gas from the expansion valve to the suction pipe.

Active circulation of brine in the freezing tank is another factor in successful ice making. The more active the circulation of the brine, the more readily it will part with its heat to the ammonia gas in the coils, and the nearer the temperature of the whole volume of the brine can be maintained at a uniform degree throughout the tank. The practical ice making engineer appreciates the value of active brine circulation, and to secure it is one of his fixed purposes. When the practical man sees "female" ice (ice that is slow in closing at the top of the block), he looks for sluggish brine circulation, and when he gets the brine to circulating regularly and actively the "female" ice disappears.

All kinds of devices are used to secure circulation, or agitation, in freezing tanks. Some

use ordinary steam pumps, others use propeller wheels; yet others, paddles or agitators. My own preference is for the centrifugal pump, with a system of brine suction and discharge pipes located inside of the tank to take



MORRIS CENTRIFUGAL PUMP WITH ENGINE.

out the suction and return the discharge brine at regular intervals of space throughout the length of the tank, thereby securing uniform motion of the brine all around the molds and coils. It is not sufficient to take the suction

out from one end of the tank and return the discharge at the opposite end of the tank, for the reason that the current of brine will flow through the channels offering the least resistance to the flow, and in consequence there will be spots entirely devoid of circulation. The circulation should be distributed equally all over the tank, and every gallon of the brine kept in active circulation—on the move all of the time. Get large circulating pumps, put in distributing suction and discharge pipes, with openings not exceeding the area of the pipes, and run the pumps to high speed, and you will have active circulation. A centrifugal pump will circulate more brine in a given time than any other known device.*

The insulation of freezing tanks is a matter of importance to the practical results. In the first place, a freezing tank should be put on a good, solid foundation—a platform set on stone or brick piers built up from hard pan. The platform should be constructed of planks across the piers, with two thicknesses of first-class waterproof building paper over the planks, then 2 × 12-inch joists set upright, with the space between the joists filled in with granulated cork rammed in close. Over the joists should be one course of rough boards and two thicknesses of paper, and then the finished and matched flooring for the tank to rest on. A bed of pitch over the flooring is a

* See Appendix for tables of capacities of centrifugal pumps.

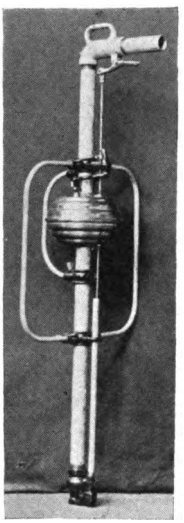
good thing to drop the tank into, and it will prevent moisture from getting under the tank and rusting it out. The platform should be made at least two feet longer and wider than the tank; and when the tank is placed in position, 2 × 12-inch joists can be run up from the platform to the top of the tank, and rough boards nailed to the joists, and two courses of building paper put on the rough boards; and the whole finished with dressed and matched flooring. The space between the casing and the tank should be filled with granulated cork well rammed in. With such insulation, the radiation of heat from the outside to the brine in the tank is reduced to a minimum. Some parties use sawdust or locomotive breeze for insulating freezing tanks. Either is considerably cheaper than granulated cork; but if either should happen to get wet from overflowing of the freezing tank or any other cause, it will be found very much dearer than the cork. The moisture will gradually drain out from the cork; but it will make either sawdust or breeze a soggy mass that will never do for insulation after having been once wet. The best is always the cheapest in the long run. *Use cork.*

Always put an overflow connection on a freezing tank, so that in case of carelessness on the part of the ice pullers, when filling the molds, if they let the molds overflow into the brine and raise the level of the brine in the

tank, the brine will not overflow into the insulation around the tank. A good place to have the overflow connection is on the front of the tank nearest to the machine, where in case of an overflow the engineer will see the wet on his floor. His remarks on such occasions are calculated to make a lasting impression on the minds of the ice pullers and cause them to be more attentive to the filling tube. "I've been there!"

A device frequently used and highly recommended by the users to prevent overflowing of the cans when being filled by careless ice pullers is the automatic can filler. This apparatus enables the ice puller to leave the can when it is being filled, as an automatic float attachment shuts off the water when the can is filled to a required height. The automatic filler further establishes a uniformity in the amount of water in each can, and consequently insures uniformity in the weight of the product from each can, a feature that will be recognized as desirable. The cuts shown on next page illustrate an automatic can filler that is giving universal satisfaction.

A purge connection to the lower manifold in a freezing tank is often a very handy thing to have. If a weak liquid, oil or any other foreign substance gets into the coils and manifolds, the purge connection comes in play very handily, and a purge valve is often a very convenient indicator of the condition of affairs

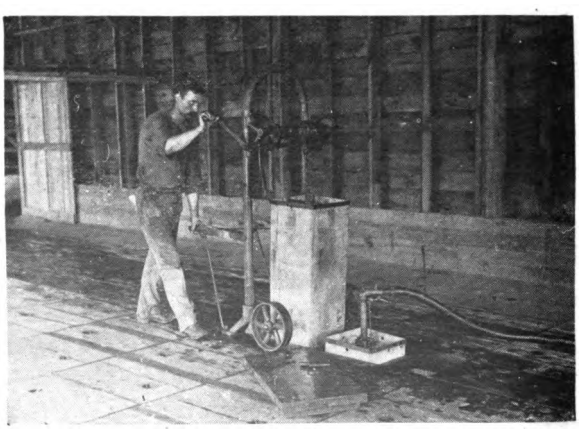


SAULS' CAN FILLER.

inside of the system when an engineer is hunting to find out what is wrong.

A standard thermometer should be placed on the brine circulating pump with the thermometer bulb down in the brine, so that the engineer can often see "where he is at" without lifting off a tank cover and getting wet with cold brine, monkeying with a tin thermometer that will not register correctly anyway.

Some engineers contend that it is necessary to have the brine kept as strong as it



SAULS BROS.' CAN FILLER IN OPERATION.

is possible to make it. That is wrong. So long as the brine is strong enough to prevent freezing around the coils and manifolds, it is quite strong enough for all practical purposes, and will answer as well, if not better, as a medium for the transmission of heat as a saturated solution would. Above a certain strength, brine will precipitate the salt at low temperatures, and a body of salt in the bottom of a freezing tank is an objectionable feature, especially if it happens to locate in a heap where you want to put a can in the tank. I have known salt to cake around the bottom of the cans and stick so tight that it was almost impossible to pull the can out of the tank without tearing the can. That was one result of over-salting the brine; and I know of no benefit to be derived from too strong brine.

I have seen all kinds of hoists used for lifting the ice cans out of freezing tanks, but the best thing for the purpose that I have ever seen is the "pneumatic" hoist. It consists of an air pump and receiver, similar in construction to the regulation air brake device used on locomotives, with an air pipe led over near the center of the traveling crane rails and an air hose from the pipe to the hoisting cylinder. The hoisting cylinder is from four inches to six inches in diameter, and about five feet long, and is suspended on a trolley which runs on the crane irons. From thirty to fifty pounds of air pressure is carried on the receiver, the pump automatically keeping the pressure at any desired point. To lift the can out of the

tank it is only necessary to turn a lever cock on the cylinder, when the piston rod will run down until the hook at the end of the rod can be engaged in the can lifter, then by reversing the cock the air is admitted under the piston, and the can, ice and all, will be hoisted up out of the tank, as fast or slow as you please, depending upon the amount of air you allow to enter the cylinder, the cock being arranged for the finest adjustment of the air admission. I have visited a factory where two 300-pound cans are taken out of the tank at once with this device, and with eight 30-ton tanks in the factory the ice is pulled with one-half the labor that was needed when the old system of hoists was in use. The device is a little expensive in first cost when compared with chain hoists or some of the other devices in vogue for the purpose, but the "pneumatic" hoist is bound eventually to find its way to the front for an ice-hoist on freezing tanks. It is like the Kodac: "You press the button and we [the air in this instance] do the rest." *Get a pneumatic hoist.*

The variety of appliances that have been tried for removing ice from the cans would take a book to describe. It is noticeable, however, that in the majority of instances the experimenter drifts back to the thawing tank as the most practical appliance for the purpose, and the majority of factories are now using the thawing tank exclusively, although some sprinkling devices are used in connection with some late ice plants.

CHAPTER XV.

DISTILLATION OF WATER.

PRACTICALLY speaking, the process of distilling water for ice making should begin at the source of the water supply. In the majority of cases the water supply is contaminated by mineral, vegetable and in some instances animal impurities. Unless these impurities are removed from the water before it is fed into the boilers, decomposition is the natural sequence, and foul gases and frequently slimy solid matter permeate the entire distilling apparatus as a consequence. The water supply should, therefore, be subjected to chemical analysis; and here the services of a chemist should be obtained, both to determine the nature of the water and to recommend the best means for the purification of the supply. The average engineer devotes very little time to chemical questions, and in consequence is hardly in a position to handle the water question to the best advantage. The chemist is, therefore, the person to consult and advise with upon this important matter.

If the boilers can be supplied with *pure water*, the greatest difficulties of distillation

will be obviated on the start, and the greatest economy of fuel consumption will result; provided, of course, that the boilers are of approved design and properly set.

Where the impurities in the water supply consist entirely of matter held in suspension, mechanical filtration will remove the impurities; but where the impurities are held in solution in the water, filtration alone will not remove the impurities, and the supply must be subjected to chemical reaction, or coagulation, before filtration to remove the impurities. Various appliances are used to purify the water supply to boilers, among the most notable being the coagulating filters, the live steam separators and purifiers, galvanic separators, alum settling tanks and gravel tanks on the back walls of boilers, exposing the water supply to the intense heat of the combustion chambers of the boiler setting. Some engineers prefer to purify the water by means of circulating devices after the supply has been fed into the boilers; and while I am free to confess that these appliances are doubtless better than to do without any purifiers, yet I believe the best plan is to *purify the water before it enters the boilers*, avoiding thereby the possibility of decomposing the impurities and the resulting foul gases entering the distilling apparatus.

The economy of an ice making plant, in so far as fuel consumption is concerned (where

the ice is to be made from distilled water), necessarily hinges upon the question of the evaporating capacity of the steam boilers. If the boilers will only evaporate from four to five pounds of water per pound of fuel consumed it would be unreasonable to expect to get from six to seven pounds of distilled water per pound of fuel consumed, for the reason that it is only the water evaporated that we can condense to produce the distilled water, and out of this we must make due allowance for waste from condensation in the pipe system, blowing off at the air exhaust on the steam condenser, and skimming the impurities from the surface of the skimmer. Clean boilers are essential to good results in evaporation, and a pure water supply will go a long way toward keeping the boilers clean and in condition to produce the best results of evaporation.

While speaking on the boiler subject, I would recommend that the boilers should always be at least 50 per cent larger than is absolutely necessary for the operation of the plant; large grate and heating surfaces will pay for themselves many times over in the operation of an ice plant. I would further recommend duplicate boiler plants, each of sufficient capacity to operate the machinery without crowding, so as to always have a reserve boiler to fire up when necessary to clean the boiler in use or to make repairs on the

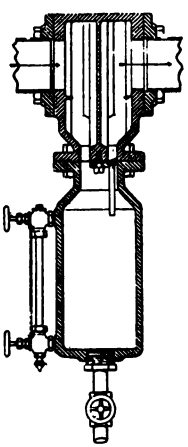
same. The utility of the duplicate boiler will be appreciated by every practical engineer, and will be demanded by every one who has had any extended experience in the business.

The pages of *Ice and Refrigeration* teem with advertisements of all kinds of devices for the purification of feed water and economizing devices to be used in connection with steam boilers, such as, for instance, the Obenchain, Myers, Cochrane, Stilwell and Hoppes cleaners; McClave grate and blower, etc., and any ice manufacturer who is having trouble with impure water supply or insufficient evaporation in the boilers would do well to investigate the merits of these devices. Many times they can get a guarantee of relief from their troubles from the manufacturers of these devices, or have the devices put in on trial, subject to acceptance only if they prove satisfactory. Almost any of the manufacturers will at once say whether they can or cannot cure the difficulty, if a report of an analysis of the feed water is submitted to them, and an analysis of the water supply should be one of the first moves to be made by an ice manufacturer when he meets with trouble in the quality of his product. The aim should be to secure a pure water supply, which when distilled will produce pure ice.

The latest accepted distilling apparatus used in connection with ice making and distilled water plants consists of an oil separa-

tor, a steam condenser, a reboiler, a skimmer, a cooling coil, one or more filters charged with sugar maple charcoal, and a cold water reservoir. To these are sometimes added a steam filter and one or more cold filters, the latter located between the cold water reservoir and the mold filling device.

The oil separator is a device such as is described in the advertisements of the Harrison Safety Boiler Works and the Austin Separator Co., in *Ice and Refrigeration*. The device is placed on the exhaust pipe, in any convenient position between the engine cylinder and the steam condenser. Its purpose is to eliminate from the exhaust steam any oil that passes through the engine cylinder, and also to relieve the exhaust line of water produced by condensation. Dead water in an exhaust pipe is frequently the cause of a bad taste in the ice, and in consequence the exhaust pipe should be drained of all condensation, and kept as dry and clean as possible.



COCHRAN OIL SEPARATOR.

The exhaust steam after passing the oil separator should be utilized as far as possible for *heating* the boiler feed water. To accomplish this end, a feed water heater can be placed between the oil separator and the steam

condenser, and the overflow water from the steam condenser passed through the heater to be heated by the exhaust steam to the highest possible temperature *without condensing the exhaust steam.* Condensation of exhaust in the heater should be avoided as far as possible, as all condensation in the heater goes to waste, thereby curtailing the product of distilled water.

To describe a steam condenser as used in connection with ice making plants at this time would be describing a multitude of different designs and patterns of *temperature exchangers*, from the simple surface condenser, made of ordinary screwed bend pipe coils, to the most complicated arrangement of apparatus imaginable. I will not, therefore, undertake to describe the construction of any one particular steam condenser, but will advise my readers to use only *surface condensers*, and to avoid any condensing apparatus which brings the exhaust steam in direct contact with the condensing water, such as the "open heater" construction of condensers. The aim is to purify the water for freezing purposes by evaporating the water to steam and condensing, reboiling and filtering the product, securing absolutely pure water. It is obvious that you cannot expose your exhaust steam to a body less pure than is the steam, and expect not to contaminate the steam by the process. In ninety-nine cases out of 100 the water used

for condensing the steam is impure water; in every case it is less pure than the steam. It is injudicious, to say the least, to mix the steam with the condensing water. With a surface condenser the steam and condensing water are never brought in direct contact; the steam is inside of the pipes, the water outside; or *vice versa* (when heater pattern of surface condensers is used). The construction of a steam condenser should be suitably arranged for surrounding conditions. Where the condensing water is reasonably clean and free from lime, the atmospheric pattern of condenser can be used to advantage; but when the condensing water is impregnated with lime or other scale producing minerals, the submerged pattern of condenser will be found the most desirable for the reason that the submerged pipes will not collect the scale as rapidly, nor will the scale that is collected become as hard and difficult to remove from the pipes as when the condenser pipes are exposed to the atmosphere; and it is a well known fact that scale on the condenser pipes detracts from the heat transmitting capacity of the condenser, and produces waste.

By experience I have found that fifty feet of 1¼-inch pipe per ton of distilled water capacity per twenty-four hours is amply sufficient for a surface steam condenser, provided the pipe is kept perfectly clean and free from scale. I recommend galvanized pipe for steam

condensers, or, in fact, for any and all parts of distilling apparatus. Galvanized pipe will not rust as readily as the ordinary black pipe, and iron rust is a material that must be avoided in connection with distilled water. It will show its red trace at once when it is present. Steam condensers should be given plenty of pitch in the pipes so as to readily drain the condensed water as fast as it is formed. The condenser should be provided with an air exhaust valve at the highest point to relieve the condenser of the air that will not condense with the steam, and thereby avoid flushing the air down with the condensed water. The steam condenser should be provided with a live steam connection, direct from the boilers, to enable a supply of live steam to be carried to the condenser when the exhaust steam from the engine is not sufficient to produce the requisite amount of distilled water, and also to enable the engineer to blow out the condenser at frequent intervals with live steam at boiler pressure, cleaning out the coils thoroughly by this means. A purge connection should be placed at the lowest point of the condenser to flush the coils when blowing out with live steam.

I have lately seen a steam condenser patterned after Rankin's idea of an ammonia condenser, consisting of 1¼-inch pipe inside of a 2-inch pipe, the smaller pipe carrying the condensing water, and the space between the

small and the large pipe acting as the condensing chamber. With this form of condenser, the condensing surface can be largely increased, as when desired water can be trickled over the outside of the 2-inch pipe and both the 1¼ and the 2-inch pipe surfaces can be utilized for heat transmitting surfaces, or either surface can be brought into play independently of the other if desired. In my opinion this form of condenser will do more work with a given quantity of water than any other form of condenser extant; for the following reasons: *First*, the steam will be spread out in the narrow space between the two pipes in the most convenient manner for readily parting with the heat; and *second*, the two pipes will necessarily expose the greatest possible heat transmitting surface that can be exposed in comparison to the volume of contents of the condenser. These two points should recommend the condenser to any thinking man. You can get the condenser from Westerlin & Campbell, who advertise in *Ice and Refrigeration*. I have just bought two 60-ton condensers from them myself.

The reboiler is merely a tank, or cylinder, usually of galvanized iron, with a steam coil in the bottom, through which live steam is circulated. The condensed water from the steam condenser is drained directly into the reboiler, where it is heated up to boiling point by the steam coil. Foul gases from the boiler and air

from the steam condenser will part company from the water in this part of the apparatus; light impurities will also rise to the surface and remain there. The reboiler is the cheapest part of a distilling apparatus to construct, and yet many plants are without one. It will pay for itself on any plant.

The skimmer is fully described by its name. It is merely for the purpose of skimming light impurities from the surface of the water, taking off oil, for instance. The skimming is accomplished by so checking the flow of distilled water from the skimmer to the cooling coil that a small amount of water will constantly overflow from the skimming chamber. A cylinder twelve inches in diameter, and from six to ten feet long, with an overflow connection at or near the top and a flushing connection in the bottom, is all that is required for a skimmer. The skimmer should be connected to the reboiler by a straight pipe, about one-half way down from the top of the reboiler and the top of the skimmer, set on a level with the point at which it is desired to carry the water level in the reboiler. As the two are connected together, the water levels will necessarily be the same in both.

The cooling coil is usually constructed of galvanized pipe, with ordinary screwed return bends—merely a flat coil with a water trough at the top and a water catch pan below. From 1¼-inch to 2-inch pipe is generally used, and

the coils are made as long as the ordinary length of stock pipe will permit, say, from sixteen to twenty feet long, and generally from sixteen to twenty pipes high. No particular rule is followed in relation to surface for a cooling coil; but I am safe in recommending fifty feet of 1¼-inch pipe as amply large for the purpose per ton of distilled water per twenty-four hours. The water is fed into the bottom of the cooling coil and taken out at the top. The cooling water flows down over the coil from top to bottom, making the most practical interchange of temperature between the two bodies of water. The cooling coil should be provided with a live steam connection at the top and a purge connection at the bottom, so that by simply closing off the water connections and opening the steam connections, the coil can be flushed out with live steam at boiler pressure; and the flushing should be done frequently if the coil is to be kept clean.

The filter is usually a galvanized iron cylinder with a tight cover and a false bottom located about a foot above the bottom of the cylinder. The filter is filled with sugar maple charcoal or crushed quartz, preferably the charcoal, and the water is led from the top of the cooling coil to the bottom of the filter and up through the bed of filtering material and out at the top. I invariably advise duplicate filters, each of large capacity, sufficient to hold at least fifty bushels of charcoal each, cross-

connected, so that either one or both of the filters can be operated independently or together, and so that one can be cut out for cleaning or recharging at any time without interfering with the operation of the other. I have tried all kinds of charcoal in filters, and have gone back to sugar maple coal, because I have found sugar maple coal to be the only coal that will be uniform every time. If you want to get into a peck of trouble with your customers on account of a "taste in your ice," you need only charge your filters with some cheap charcoal, and I will guarantee a "taste" that will cling to your memory after you have lost your hearing and sight. Twenty-five dollars' worth of cheap charcoal cost me two hundred dollars and a red hot cussing in my youth, and I know better now. The filters should be provided with purge connections at the bottom, and should be washed out frequently. Opening the purge connection will wash down through the bed of filtering material and carry off the greater part of the impurities collected. A new charge of filtering material should be put in the filters at least once a season, however; it will get foul after a time, and when foul it is only fit for fuel.

The cold water reservoir, sometimes called "fore-cooler," consists of a reservoir containing a system of coils through which the ammonia gas from the freezing tanks is passed before the gas is delivered to the compressors.

Its purpose is to cool the distilled water by means of the excess duty in the ammonia gas from the freezing tanks, and to deliver the water to the molds as near the freezing point as possible. With a good apparatus it is possible to fore-cool the water to an average of 40° F. in this manner. (I have seen distilled water in a reservoir at 36°.) There are two points to be considered in the construction of a cold water reservoir: first, the size of the suction pipe to the compressors, as upon the area of this pipe depends the number of coils that can be placed in the reservoir, and the coilage area should equal the suction pipe area to get the best results; and second, the number of feet of surface required to lower the temperature of the water from its temperature when admitted into the reservoir to the temperature at which it is desired to deliver the water to the molds. Having decided these points, the reservoir can be built to suit the coils. It is not good policy to build the reservoir and then construct the coils to suit the reservoir, as a friend of mine did recently. I quote from a letter I wrote to him on the subject as follows:

"Your fore-cooler (cold water reservoir) is just as much a factor for the transmission of heat as your evaporating or expansion surfaces are, and the proportions of surface should be in proportion to the amount of heat to be transmitted in each case. To illustrate

the point, let us take water at say 80° F. and convert it into ice at 14° F. (the temperature at which the ice is usually taken from the freezing tank), and charge the proportions of heat transmitted to the proper part of the apparatus in each instance. First, we fore-cool the water from 80° to say 40° F. (theoretically it should be fore-cooled to 32° F., but it is unnecessary for me to tell you that practice does not reach the perfection of theory); our fore-cooler, therefore, would be called upon to transmit 40 B. T. units of heat for each pound of water. Now, the water is delivered to the freezing tank at 40° F., and we first cool the water down to 32° F., which transmits 8 B. T. units; then to convert water at 32° F. to ice at 32° F., say 143 B. T. units; then ice from 32° F. to 14°, say 18 B. T. units, and we have a freezing total of say 169 B. T. units. (Note that I do not make any allowance for the difference in specific heat of ice and water, but in this calculation the same would not be material to the total result of the illustration.) The only conclusion to be drawn from this calculation is that the fore-cooler, to be correct, should expose at least 25 per cent as much cooling surface as is exposed in the freezing tank, if it is expected to do its proportion of the work of transmitting the heat from the water."

The above quotation from the letter tells the cold water reservoir story as it should be,

which I regret to say is not the way the reservoirs will be found in the majority of plants in this country. The reservoir should be provided with means for cleaning it out handily, and it should be cleaned out at regular intervals, as it acts as a settling chamber and necessarily is the most exposed portion of the distilling apparatus.

In starting up a distilling apparatus care should be exercised to see that every portion of the apparatus is thoroughly cleansed before steam is admitted for practical distillation. One of the easiest means of cleaning out the apparatus is to wash it out with a solution of soda ash, allowing the condenser, cooling coil, reboiler and skimmer to stand full of the solution for some hours, and then drain and blow out thoroughly with live steam until all trace of the solution has disappeared. The solution will cut all grease in the pipes and chambers, and the steam will clean the apparatus thoroughly.

When starting up a plant in the spring, this system of cleaning out should be well attended to, especially if the apparatus has been drained out during the season when the plant has been shut down. Cleanliness is next to godliness, and pure distilled water is expected to be the acme of cleanliness by the parties who are paying for the ice and the water.

APPENDIX.

SOURCES OF COLD.*

THE production of cold—the abstraction of heat—is a curious subject of inquiry. When a salt is dissolved in water, cold is produced. When a liquid vaporizes, the heat, latent and sensible, necessary for the production of the vapor, is abstracted from some other body in contact with the liquid, and cold is produced. When spirits of wine or aromatic vinegar, for example, is thrown on the body, a sense of cold immediately results from the vaporization of the liquid, which draws heat from the body. If air is allowed to expand, there is a reduction of temperature, and a translation of heat from neighboring bodies. Again, in hot climates, water is successfully cooled in porous vessels, through the pores of which the water passes to the exterior and is vaporized, and the cooling process is accelerated by a current of air directed upon the vessel, which quickens the vaporization.

Siebe's ice making machine, invented ori-

*Clark's Manual.

ginally by Jacob Perkins in 1834, is based on the principle of producing cold by the evaporation of a volatile fluid—ether by preference. The fluid is placed in an air tight vessel, and evaporated *in vacuo*, the vacuum being formed by means of a pump, which, in its continued efforts to reduce the pressure, promotes the evaporation of the fluid at a low temperature. A temperature of 50° below the freezing point may be effected; but in place of an unprofitably low temperature, the cooling action is distributed through the mass of salt water employed as the freezing medium, the salt water retaining its fluidity below 32° F., and circulating in the refrigerator around and between the ice molds, which are filled with fresh water. The water in the molds is successively frozen, and replaced by fresh molds filled with water.

Carré's cooling apparatus is based on the fact that water when cold absorbs a large quantity of ammoniacal gas, which when the water is heated escapes, and is condensed in a cold vessel. On the contrary, when the water just heated becomes cold, a vacuum is formed and excites a rapid evaporation of the ammonia into the vessel of cooled water, when it is again absorbed. The heat necessary for the evaporation of the ammonia is extracted from the water surrounding the vessel in which the liquid ammonia is contained, and the water consequently is frozen.

FRIGORIFIC MIXTURES.

For the production of intense cold, mixtures of various salts and acids in various proportions with water are very effective. But more intense degrees of cold are produced with snow or ice.

Table No. 1 contains the ordinary mixtures for the artificial production of cold, known as freezing mixtures. The first part of the table comprises mixtures of salts and acids with each other and with water; the second part, mixtures of salts and acids with snow or ice.

The blanks in the third column of the table indicate that the thermometer sinks to the degrees named in the second column, but never lower, whatever may be the initial temperature of the materials when mixed.

The vessels containing the mixtures should be cooled before the elements are put into them.

If the materials of the mixtures enumerated in the first part of the table be mixed at a higher temperature than that given in the table, namely, 50° F., the fall of temperature is greater. Thus, if the most powerful of these mixtures, No. 11, be made at the temperature 80° F., it will sink the thermometer to + 2°, making a fall of 78°, as against 71° in the table.

The third part of the table contains frigorific mixtures partly selected from the other parts, and combined so as to extend the cold to the extreme degree —91° F. The ma-

terials should be cooled previously to being mixed to the initial temperature, by mixtures taken from previous parts of the table.

TABLE No. 1.—FRIGORIFIC MIXTURES.

FIRST PART.—Proportional mixtures of salts and acids with water.

Mixtures.	Fall of Temperature.	Degrees of Cold Produced.
	Fahrenheit.	Fahr.
1. Nitrate of ammonia..... 1 Water..... 1	from +50° to +4°	46°
2. Muriate of ammonia.... 5 Nitrate of potash..... 5 Water.....16	from +50° to +10°	40
3. Muriate of ammonia.... 5 Nitrate of potash.. ..... 5 Sulphate of soda..... 8 Water.....16	from +50° to +4°	46
4. Sulphate of soda... . .. 3 Diluted nitric acid..... 2	from +50° to −3°.....	53
5. Nitrate of ammonia.... 1 Carbonate of soda..... 1 Water..... 1	from +50° to −7°.....	57
6. Phosphate of soda ..... 9 Diluted nitric acid 4	from +50° to −12°.....	62
7. Sulphate of soda... .... 8 Hydrochloric acid..... 5	from +50° to 0°.... ...	50
8. Sulphate of soda .... 5 Dilute sulphuric acid... 4	from +50° to +3°	47
9. Sulphate of soda..... 6 Muriate of ammonia.... 4 Nitrate of potash..... 2 Dilute nitric acid..... 4	from +50° to −10°.....	60
10. Sulphate of soda ..... 6 Nitrate of ammonia.... 5 Dilute nitric acid..... 4	from +50° to −14°. ...	64
11. Phosphate of soda ..... 9 Nitrate of ammonia .... 6 Dilute nitric acid..... 4	from +50° to −21°....	71

SECOND PART.—Proportional mixtures of salts and acids with snow or ice.

Mixtures.	Fall of Temperature.	Degrees of Cold Produced.
	Fahrenheit.	Fahr.
12. Muriate of soda (common salt)..... 1 Snow, or pounded ice... 2	from any temp. to −5°	——
13. Muriate of soda ..... 2 Muriate of ammonia.... 1 Snow, or pounded ice... 5	" " " to −12°	——
14. Muriate of soda..... 10 Muriate of ammonia.... 5 Nitrate of potash.. ..... 5 Snow, or pounded ice...24	" " " to −18°	——
15. Muriate of soda..... 5 Nitrate of ammonia..... 5 Snow, or pounded ice...12	" " " to −25°	——
16. Dilute sulphuric acid... 2 Snow..... 3	from + 32° to −23°.....	55°
17. Muriatic acid..... 5 Snow..... 8	from + 32° to −27°.....	59
18. Dilute nitric acid..... 4 Snow 7	from + 32° to −30°.....	62
19. Muriate of lime..... 5 Snow 4	from + 32° to −40°.....	72
20. Crystallized muriate of lime..... 3 Snow..... 2	from + 32° to −50°.....	82
21. Potash..... 4 Snow... 3	from + 32° to −51°.....	83

THIRD PART.—Mixtures partly selected from the foregoing series, and combined so as to increase or extend the cold to the greatest extremes.

Mixtures.	Fall of Temperature.	Degrees of Cold Produced.
	Fahrenheit.	Fahr.
22. Sea salt..... 5 Muriate of ammonia } .. 5 Nitrate of potash } Snow, or pounded ice... 1	from −5° to −18°.....	13°
23. Sea salt..... 5 Nitrate of ammonia .... 5 Snow, or pounded ice... 12	from −18° to −25°....	7
24. Phosphate of soda..... 5 Nitrate of ammonia.... 3 Dilute nitric acid..... 4	from 0° to −34°.....	34
25. Phosphate of soda..... 3 Nitrate of ammonia .... 2 Dilute mixed acids. 4	from −34° to −50°.....	16

THIRD PART—Concluded.

Mixtures.	Fall of Temperature.	Degrees of Cold Produced.
	Fahrenheit.	Fahr.
26. Snow..... 3 Dilute nitric acid..... 2	from 0° to —46°.....	46
27. Snow..... 8 Dilute sulphuric acid... 3 Dilute nitric acid..... 3	from —10° to —56°.....	46
28. Snow..... 1 Dilute sulphuric acid... 1	from —10° to —60°.....	50
29. Snow..... 3 Muriate of lime..... 4	from + 20° to —48°.....	68
30. Snow..... 3 Muriate of lime.. 4	from + 10° to —54°.....	64
31. Snow..... 2 Muriate of lime 3	from —15° to —68°.....	53
32. Snow..... 1 Crystallized muriate of lime..... 2	from 0° to —66°.....	66
33. Snow..... 1 Crystallized muriate of lime..... 3	from —40° to —73°.....	33
34. Snow..... 8 Dilute sulphuric acid...10	from —68° to —91°.....	23

COLD BY EVAPORATION.

M. Gay-Lussac directed a current of air, dried, or desiccated, by being passed through chloride of calcium, upon the bulb of a thermometer wrapped in moist cambric. The temperature was lowered from 10° to 26° F., according to the temperature of the current of air, which varied from 32° to 77° F. It is presumed that the surrounding temperature was the same as that of the current. The following are the falls of temperature for currents of air of given temperatures:

Temperature of current, Fahr., 32°, 41°, 50°, 59°, 68°, 77°.
Fall of temperature, " 10.5°, 13°, 16°, 19.5°, 23°, 26.5°.

The most intense cold as yet known was produced by Professor Faraday in the course of his experiments on the liquefaction and

solidification of gases, from the evaporation of a mixture of solid carbonic acid and sulphuric ether under the receiver of an air pump. For the following pressures, measured in inches of mercury, and given also in pounds per square inch, he obtained the corresponding temperatures subjoined:

Inches of mercury	28.4,	19.4,	9.6,	7.4,	5.4,	3.4,	2.4,	1 4,	1.2
Pounds per square inch.	14.0,	9.5,	4.6,	3.6,	2.7,	1.7,	1.2,	0.7,	0.6
Temperatures, Fahr.....	-107°,	-112°,	-121°,	-125°,	-132°,	-139,	-146°,	-161°,	-166°

Showing that when a perfect vacuum was nearly approached, an intense cold, measured by —166° F., was attained by the evaporation of a mixture of solid carbonic acid and sulphuric ether.

COOLING OF HOT WORT BY COLD WATER IN METALLIC REFRIGERATORS.

From the instructive discussion of the principles of brewery engineering in *Engineering*, Vol. VI, the following particulars are derived of the performance of tubular refrigerators, in which cold water is passed through thin metallic tubes, which are surrounded by the wort to be cooled. The water and the wort are moved in opposite directions in such a manner that the cold water, on its entrance into the refrigerator, meets the cooled wort just before it leaves the refrigerators, and the warmed water passes away from the refrigerator where the hot wort enters. The following are particulars of the performance in five experiments:

TABLE NO. 2.—RESULTS OF PERFORMANCE OF METALLIC REFRIGERATORS IN COOLING HOT WORT WITH COLD WATER.

Area of Cooling Surface of Refrigerator.	Wort.					Water.			
	Specific Gravity.	Quantity Passed through Per Hour.	Initial Temperature.	Final Temperature.	Cooled Down.	Quantity Passed through Per Hour.	Initial Temperature.	Final Temperature.	Warmed up.
Sq. Feet.		Bbls.	Fahr	Fahr	Fahr	Bbls.	Fahr	Fahr	Fahr
1. 881	.....	33.9	212°	72°	140°	61.1	65°	169°	104°
2. 514	1.104	36.1	155	59	96	75.5	54	100	46
3. 514	1.188	36.6	191	59	132	99.5	54	100	46
4. 514	1.035	47.3	193	59	134	90.7	54	100	46
5. 514	1.018	48.0	178	59	119	102.0	54	100	46

NOTE 1.—A barrel contains thirty-six gallons, or 360 pounds of water.

2.—The temperature of the air in Nos. 2 and 4 was 44° F., and in Nos. 3 and 5, 40°.

Dealing with the data of this table, the following are the mean temperatures and differences of temperature of the wort and the water, with the quantities of heat transmitted per unit of surface, temperature and time:

Number of Experiment.	Mean Temperatures.		Mean Difference of Temperature.	Heat Transmitted per Square Foot per Degree per Hour.	
	Of Wort.	Of Water.		Measured by Reduction of Temperature of Wort.	Measured by Increase of Temperature of Water.
	Fahr.	Fahr.	Fahr.	Units.	Units.
1	142°	117°	25°	78	104
2	107	77	30	81	81
3	125	77	48	71	67
4	126	77	49	91	59
5	118.5	77	41.5	96	79
Averages.....	.....	.....	.....	83.4	78

ANHYDROUS AMMONIA.

Ammonia is a compound of one volume of nitrogen with three volumes of hydrogen, and is therefore represented by the chemical formula NH_3. It contains by weight 82.35 per cent nitrogen and 17.65 per cent hydrogen. Its molecular weight is 17.

Ammonia is a colorless gas possessing a very characteristic pungent smell. It is much lighter than air, having a specific gravity (air 1) of 0.586, one liter of gas weighing, at the normal temperature and pressure, 0.76193 grams. By mechanical pressure and cooling, it is converted from a gaseous to a liquid state (liquid anhydrous ammonia) which boils under the ordinary atmospheric pressure at $28\frac{6}{10}^{\circ}$ below zero, or $240\frac{1}{2}^{\circ}$ lower than the boiling point of water under the same conditions. One pound of the liquid at 32° will occupy 21.017 cubic feet of space when evaporated at the atmospheric pressure. The specific heat of ammonia gas, as determined by Regnault (capacity for heat), is 0.50836. Its latent heat of evaporation is about 560 thermal units at 32° Fahrenheit, at which temperature one pound of the liquid, evaporated under a pressure of 15 pounds per square inch, will occupy 21 cubic feet.

TESTING ANHYDROUS AMMONIA.

Usually ammonia manufacturers sell their goods subject to the condition and agreement, on the part of the purchaser, that a sample be drawn from each cylinder upon arrival and subjected to a test before emptying the contents, no reclamation being allowed on account of deficiency in quality or strength after a cylinder has been emptied or partly emptied. Therefore it is important that the consumer satisfy himself of the purity of the ammonia before drawing off the contents of the cylinder.

EVAPORATION TEST.

Any dealer in chemical supplies will furnish an 8-oz. flat bottom, wide neck, Bohemian glass boiling flask (in case of breakage it is well to have several of these). Fit in the neck a stopper having a ¼-inch vent hole punctured through for escape of the gas. Insert in this hole a short glass tube. Procure a piece of ⅜-inch iron pipe threaded at one end; bend the pipe to such a shape that the threaded end can be connected with the cylinder valve; put the wrench on the valve of the cylinder and open it gently; allow a little of the ammonia gas to escape at first in order to purge the pipe and valve, then draw into the test flask from 2½ to 4 ounces of the liquid ammonia. When this is accomplished, remove the test flask at once, and insert in the neck the stopper with vent tube, then place it in such a position as will allow a small stream of water to flow over the sides of the flask. Under these conditions the ammonia will boil quickly and soon evaporate. Any residue remaining in the flask indicates impurities. Care is necessary in drawing off the sample, as a very little moisture in the test flask or in the pipe, or a brief exposure to the atmosphere, will at once affect it.

AQUA AMMONIA.

The aqueous solution of ammonia, known as aqua ammonia, liquid ammonia, etc., is ammonia gas absorbed by water. The solution possesses the characteristic odor of the gas; colors red litmus paper blue, and turmeric paper brown, and possesses all the characteristic properties of an alkali. The strength of aqua ammonia is approximately determined by specific gravity scales or hydrometers, those of Beaumé being usually employed for the purpose. When ammonia dissolves in water a considerable evolution of heat takes place, the volume of the

solution increasing at the same time. The solubility of ammonia in water at various temperatures is given in the following table:

One vol. of water at	32°	Fahrenheit	absorbs	1,050	vols.	ammonia
"	"	"	"	"	50°	"	"	813	"	"
"	"	"	"	"	59°	"	"	727	"	"
"	"	"	"	"	68°	"	"	654	"	"

TESTS FOR IMPURITIES IN AQUA AMMONIA.

CARBONATES.

Take a sample of aqua ammonia, dilute with two or three parts distilled water, add clear lime water. If carbonates are present, a white precipitate of carbonate of lime will form.

SULPHATES.

Neutralize a sample of aqua ammonia with chemically pure muriatic acid. If white precipitate is noticed, on addition of barium chloride, sulphates are present.

CHLORIDES.

Neutralize sample of aqua ammonia with chemically pure nitric acid, and add a few drops of solution of silver nitrate. A white precipitate indicates presence of chlorides.

ORGANIC MATTER.

Dilute sample of aqua ammonia with one part distilled water, and add chemically pure nitric acid until the latter is in slight excess. Any impurities of an empyreumatic nature will give a pink color.

PROF. LUNGE'S TABLE SHOWING SPECIFIC GRAVITY AND STRENGTH OF AQUA AMMONIA AT 59° FAHRENHEIT.

°Beaumé	Sp. Gr.	Per Ct. Am.Gas	°Beaumé	Sp. Gr.	Per Ct. Am.Gas
10	1.000	0.00	21	0.930	18.64
10.5	0.997	0.80	21.5	0 927	19.56
11	0.993	1.60	22	0.924	20.49
12	0.986	3.20	22.5	0.921	21.44
13	0.979	5.05	23	0.918	22.39
14	0.973	6.56	23.5	0.915	23.36
15	0.967	8.08	24	0.912	24.33
16	0.960	9.91	24.5	0.909	25.32
17	0.954	11.60	25	0.906	26.31
18	0.948	13.31	25.5	0.903	27.32
18.5	0.945	14.18	26	0.900	28.33
19	0.942	15.05	26.5	0.897	29.35
20	0.936	16.82	27	0.894	30.37
20.5	0.933	17.73	.....	... .	.....

PROPERTIES OF SOLUTION OF CHLORIDE OF CALCIUM.

Percentage by Weight.	Specific Heat.	Spec. Grav. at 60° F.	Freezi'g Pt. Degrees F.	Freezi'g Pt. Degs. Cels.
1	0.996	1.009	31	— 0.5
5	0.964	1.043	27.5	— 2.5
10	0.896	1.087	22	— 5.6
15	0.860	1.134	15	— 9.6
20	0.834	1.182	5	—14.8
25	0.790	1.234	8	—22.1

TABLE SHOWING PROPERTIES OF SOLUTION OF SALT.

(Chloride of Sodium.)

1 Percentage of Salt by Weight.	2 Pounds of Salt per Gallon of Solution.	3 Degrees on Salometer at 60° F.	4 Weight per Gallon at 39° F.-4° C.	5 Specific Gravity at 39° F.-4° C.	6 Specific Heat.	7 Freezing Point, Fahrenheit.
1	0.084	4	8.40	1.007	0.992	30.5
2	0.169	8	8.46	1.015	.....	29.3
2.5	0.212	10	8.50	1.019	.....	28.6
3	0.256	12	8.53	1.023	.....	27.8
3.5	0.300	14	8.56	1.026	.....	27.1
4	0.344	16	8.59	1.030	.....	26.6
5	0.433	20	8.65	1.037	0.960	25.2
6	0.523	24	8.72	1.045	.....	23.9
7	0.617	28	8.78	1.053	.....	22.5
8	0.708	32	8.85	1.061	.....	21.2
9	0.802	36	8.91	1.068	.....	19.9
10	0.897	40	8.97	1.076	0.892	18.7
12	1.092	48	9.10	1.091	.....	16.0
15	1.389	60	9.26	1.115	0.855	12.2
20	1.928	80	9.64	1.155	0.829	6.1
24	2.376	96	9.90	1.187	.....	1.2
25	2.488	100	9.97	1.196	0.783	.5
26	2.610	.....	10.04	1.204	.....	—1.1
29	.....	.....	.....	.....	.....	—4.7

To determine the weight of one cubic foot of brine, multiply the values given in column 4 by 7.48.

To determine the weight of salt to one cubic foot of brine, multiply the values given in column 2 by 7.48.

WOOD'S TABLE OF SATURATED AMMONIA.*
Recalculated by GEORGE DAVIDSON, M.E.

Temperature.		Pressure, Absolute.								Temperature.
Degrees F. t.	Absolute. T.	Pounds per Sq. Foot. P.	Pounds per Sq. Inch. p.	Gauge Pressure, Pound per Sq. Inch.	Heat of Vaporization, Thermal Units. h.	Volume of Vapor per Pound, Cubic Feet. v.	Volume of Liquid per Pound. Cubic Feet. v_1	Weight of Vapor in Pounds per Cubic Foot. w	Weight of Liquid in Pounds per Cubic Foot. w_1	Degrees F.
−40	420.66	1539.90	10.69	−4.01	579.67	24.388	.02348	.0410	42.589	−40
39	1	1584.43	11.00	−3.70	579.07	23.735	.02351	.0421	42.535	39
38	2	1630.03	11.32	−3.38	578.42	23.102	.02354	.0433	42.483	38
37	3	1676.71	11.64	−3.06	577.88	22.488	.02357	.0444	42.427	37
36	4	1724.51	11.98	−2.72	577.27	21.895	.02359	.0457	42.391	36
−35	425.66	1773.43	12.31	−2.39	576.68	21.321	.02362	.0469	42.337	−35
34	6	1823.50	12.66	−2.04	576.08	20.763	.02364	.0482	42.301	34
33	7	1874.73	13.02	−1.68	575.48	20.221	.02366	.0495	42.265	33
32	8	1927.17	13.38	−1.32	574.89	19.708	.02368	.0507	42.213	32
31	9	1980.78	13.75	−0.95	574.39	19.204	.02371	.0521	42.176	31
−30	430.66	2035.69	14.13	−0.57	573.69	18.693	.02374	.0535	42.123	−30
29	1	2091.83	14.53	−0.17	573.08	18.225	.02378	.0549	42.052	29
28	2	2149.23	14.92	+0.22	572.48	17.759	.02381	.0563	42.000	28
27	3	2207.94	15.33	+0.63	571.89	17.307	.02384	.0577	41.946	27
26	4	2267.97	15.75	+1.05	571.28	16.869	.02387	.0593	41.893	26
−25	435.66	2329.34	16.17	+1.47	570.68	16.446	.02389	.0608	41.858	−25
24	6	2392.09	16.61	1.91	570.08	16.034	.02392	.0624	41.806	24
23	7	2456.23	17.05	2.35	569.48	15.633	.02395	.0640	41.754	23
22	8	2520.45	17.50	2.8	568.88	15.252	.02398	.0656	41.701	22
21	9	2588.77	17.97	3.27	568.27	14.875	.02401	.0672	41.649	21
−20	440.66	2657.23	18.45	+3.75	567.67	14.507	.02403	.0689	41.615	−20
19	1	2727.17	18.94	4.24	567.06	14.153	.02406	.0706	41.563	19
18	2	2798.62	19.43	4.73	566.43	13.807	.02409	.0725	41.511	18
17	3	2871.61	19.94	5.24	565.85	13.475	.02411	.0742	41.480	17
16	4	2946.17	20.46	5.76	565.25	13.150	.02414	.0760	41.425	16
−15	445.66	3022.31	20.99	+6.29	564.64	12.834	.02417	.0779	41.374	−15
14	6	3100.07	21.53	6.83	564.04	12.527	.02420	.0798	41.322	14
13	7	3179.45	22.08	7.38	563.43	12.230	.02423	.0818	41.271	13
12	8	3260.52	22.64	7.94	562.82	11.939	.02425	.0838	41.237	12
11	9	3343.29	23.22	8.52	562.21	11.659	.02428	.0858	41.186	11
−10	450.66	3427.75	23.80	+9.10	561.61	11.385	.02431	.0878	41.135	−10
9	1	3513.97	24.40	9.70	560.99	11.117	.02434	.0899	41.084	9
8	2	3601.97	25.01	10.31	560.39	10.860	.02437	.0921	41.034	8
7	3	3691.75	25.64	10.94	559.78	10.604	.02439	.0943	41.000	7
6	4	3783.37	26.27	11.57	559.17	10.362	.02442	.0965	40.950	6
−5	455.66	3876.85	26.92	12.22	558.56	10.125	.02445	.0988	40.900	−5
4	6	3972.62	27.59	+12.89	557.94	9.894	.02448	.1011	40.845	4
3	7	4069.48	28.26	13.56	557.33	9.669	.02451	.1034	40.799	3
2	8	4168.70	28.95	14.25	556.73	9.449	.02454	.1058	40.749	2
1	9	4269.90	29.65	14.95	556.11	9.234	.02457	.1083	40.700	1
0	460.66	4373.10	30.37	+15.67	555.50	9.028	.02461	.1107	40.650	0
+1	1	4478.32	31.10	16.40	554.88	8.825	.02463	.1133	40.601	+1
2	2	4485.60	31.84	17.14	554.27	8.630	.02466	.1159	40.551	2
3	3	4694.96	32.60	17.90	553.65	8.436	.02469	.1186	40.502	3
4	4	4806.46	33.38	18.68	553.04	8.250	.02472	.1212	40.453	4

* For values at temperatures higher than 100° F. see Wood's table on page 201.

WOOD'S TABLE OF SATURATED AMMONIA—Continued.

Temperature.		Pressure, Absolute.								Temperature.
Degrees F. t.	Absolute. T.	Pounds per Sq. Ft. P.	Pounds per Sq. Inch p.	Gauge Pressure, Pound per Sq. Inch.	Heat of Vaporization, Thermal Units. h.	Volume of Vapor per Pound, Cubic Feet. v.	Volume of Liquid per Pound, Cubic Feet. v_1	Weight of Vapor in Pounds per Cubic Foot. w	Weight of Liquid in Pounds per Cubic Foot. w_1	Degrees F.
+5	465.66	4920.11	34.16	+19.46	552.43	8.070	.02475	.1240	40.404	+5
6	6	5035.95	34.97	20.27	551.81	7.892	.02478	.1267	40.355	6
7	7	5153.99	35.79	21.09	551.19	7.717	.02480	.1296	40.322	7
8	8	5274.28	36.63	21.93	550.58	7.553	.02483	.1324	40.274	8
9	9	5396.93	37.48	22.78	549.96	7.388	.02486	.1253	40.225	9
+10	470.66	5521.71	38.34	+23.64	549.35	7.229	.02490	.1383	40.160	+10
11	1	5649.48	39.23	24.53	548.73	7.075	.02493	.1413	40.112	11
12	2	5778.50	40.13	25.43	548.11	6.924	.02496	.1444	40.064	12
13	3	5910.52	41.04	26.34	547.49	6.786	.02499	.1474	40.016	13
14	4	6044.96	41.98	27.28	546.88	6.632	.02502	.1507	39.968	14
+15	475.66	6182.00	42.94	+28.24	546.26	6.491	.02505	.1541	39.920	+15
16	6	6321.24	43.90	29.20	545.63	6.355	.02508	.1573	39.872	16
17	7	6463.24	44.88	30.18	545.01	6.222	.02511	.1607	39.872	17
18	8	6607.77	45.89	31.19	544.39	6.093	.02514	.1641	39.777	18
19	9	6754.90	46.91	32.21	543.74	5.966	.02517	.1676	39.729	19
+20	480.66	6904.68	47.95	+33.25	543.15	5.843	.02520	.1711	39.682	+20
21	1	7057.15	49.01	34.31	542.53	5.722	.02523	.1748	39.635	21
22	2	7211.33	50.09	35.39	541.90	5.605	.02527	.1784	39.572	22
23	3	7370.27	51.18	36.48	541.28	5.488	.02529	.1822	39.541	23
24	4	7530.96	52.30	37.60	540.66	5.378	.02533	.1860	39.479	24
+25	485.66	7694.52	53.43	+38.73	540.03	5.270	.02536	.1897	39.432	+25
26	6	7860.89	54.59	39.89	539.41	5.163	.02539	.1937	39.386	26
27	7	8030.16	55.76	41.06	538.78	5.058	.02542	.1977	39.339	27
28	8	8202.38	56.96	42.26	538.16	4.960	.02545	.2016	39.292	28
29	9	8377.56	58.17	43.47	537.53	4.858	.02548	.2059	39.246	29
+30	490.66	8555.74	59.42	+44.72	536.91	4.763	.02551	.2099	39.200	+30
31	1	8736.96	60.67	45.97	536.28	4.668	.02554	.2142	39.115	31
32	2	8921.26	61.95	47.25	535.66	4.577	.02557	.2185	39.108	32
33	3	9108.71	63.25	48.55	535.03	4.486	.02561	.2229	39.047	33
34	4	9299.32	64.58	49.88	534.40	4.400	.02564	.2273	39.001	34
+35	495.66	9493.07	65.92	+51.22	533.78	4.314	.02568	.2318	38.940	+35
36	6	9690.04	67.29	52.59	533.13	4.234	.02571	.2362	38.894	36
37	7	9890.75	68.68	53.98	532.52	4.157	.02574	.2413	38.850	37
38	8	10093.91	70.09	55.39	531.89	4.068	.02578	.2458	38.789	38
39	9	10300.88	71.53	56.83	531.26	3.989	.02582	.2507	38.729	39
+40	500.66	10511.16	72.99	+58.29	530.63	3.915	.02585	.2554	38.684	+40
41	1	10724.95	74.48	59.78	529.99	3.839	.02588	.2605	38.639	41
42	2	10942.18	75.99	61.29	529.36	3.766	.02591	.2655	38.595	42
43	3	11162.93	77.52	62.82	528.73	3.695	.02594	.2706	38.550	43
44	4	11387.21	79.08	64.38	528.10	3.627	.02597	.2757	38.499	44
+45	505.66	11615.12	80.66	+65.96	527.47	3.559	.02600	.2809	38.461	+45
46	6	11846.64	82.27	67.57	526.83	3.493	.02603	.2863	38.417	46
47	7	12081.80	83.90	69.20	526.20	3.428	.02606	.2917	38.373	47
48	8	12320.71	85.56	70.86	525.57	3.362	.02609	.2974	38.328	48
49	9	12563.36	87.25	72.55	524.93	3.303	.02612	.3027	38.284	49
+50	510.66	12809.91	88.96	+74.26	524.30	3.242	.02616	.3084	38.226	+50
51	1	13060.21	90.70	76.00	523.66	3.182	.02620	.3145	38.167	51
52	2	13314.43	92.46	77.76	523.03	3.124	.02623	.3201	38.124	52
53	3	13572.52	94.25	79.55	522.39	3.069	.02626	.3258	38.080	53
54	4	13834.64	96.07	81.37	521.76	3.012	.02629	.3220	38.037	54

WOOD'S TABLE OF SATURATED AMMONIA—Continued.

Temperature.		Pressure, Absolute.		Gauge Pressure. Pound per Sq. Inch.	Heat of Vaporization, Thermal Units. h.	Volume of Vapor per Pound, Cubic Feet. v.	Volume of Liquid per Pound, Cubic Feet. v_1	Weight of Vapor in Pounds per Cubic Foot. w.	Weight of Liquid in Pounds per Cubic Foot. w_1	Temperature. Degrees F.
Degrees F. t.	Absolute. T	Pounds per Sq. Foot. P.	Pounds per Sq. Inch. p.							
+55	515.66	14100.74	97.92	+83.22	521.12	2.958	.02632	.3380	37.994	+55
56	6	14370.92	99.80	85.10	520.48	2.905	.02636	.3442	37.936	56
57	7	14645.18	101.70	87.00	519.84	2.853	.02639	.3505	37.893	57
58	8	14923.98	103.64	88.94	519.20	2.802	.02643	.3568	37.835	58
59	9	15206.28	105.60	90.90	518.57	2.753	.02646	.3632	37.793	59
+60	520.66	15493.09	107.59	+92.89	517.93	2.705	.02651	.3697	37.736	+60
61	1	15784.23	109.61	94.91	517.29	2.658	.02654	.3762	37.678	61
62	2	16079.67	111.66	96.96	516.65	2.610	.02658	.3831	37.622	62
63	3	16379.51	113.75	99.05	516.01	2.565	.02661	.3898	37.579	63
64	4	16683.75	115.86	101.16	515.37	2.520	.02665	.3968	37.523	64
+65	525.66	16992.50	118.03	+103.33	514.73	2.476	.02668	.4039	37.481	+65
66	6	17305.70	120.18	105.48	514.09	2.433	.02671	.4110	37.439	66
67	7	17623.45	122.38	107.68	513.45	2.389	.02675	.4189	37.383	67
68	8	17945.89	124.62	109.92	512.81	2.351	.02678	.4254	37.341	68
69	9	18272.81	126.89	112.19	512.16	2.310	.02682	.4329	37.285	69
+70	530.66	18604.53	129.19	+114.49	511.52	2.272	.02686	.4401	37.230	+70
71	1	18941.00	131.54	116.84	510.87	2.233	.02689	.4479	37.188	71
72	2	19282.21	133.90	119.20	510.22	2.194	.02693	.4558	37.133	72
73	3	19628.32	136.31	121.61	509.58	2.153	.02697	.4645	37.079	73
74	4	19979.22	138.74	124.04	508.93	2.122	.02700	.4712	37.037	74
+75	535.66	20335.16	141.22	+126.52	508.29	2.087	.02703	.4791	36.995	+75
76	6	20696.00	143.72	129.02	507.64	2.052	.02706	.4873	36.954	76
77	7	21061.85	146.26	131.56	506.99	2.017	.02710	.4957	36.900	77
78	8	21432.82	148.84	134.14	506.34	1.995	.02714	.5012	36.845	78
79	9	21808.85	151.45	136.75	505.69	1.952	.02717	.5123	36.805	79
+80	540.66	22190.15	154.10	+139.40	505.05	1.921	.02721	.5205	36.751	+80
81	1	22576.51	156.78	142.08	504.40	1.889	.02725	.5294	36.696	81
82	2	22968.88	159.50	144.80	503.75	1.858	.02728	.5382	36.657	82
83	3	23365.38	162.26	147.56	503.10	1.827	.02732	.5473	36.603	83
84	4	23767.81	165.05	150.35	502.45	1.799	.02736	.5558	36.549	84
+85	545.66	24175.61	167.88	+153.18	501.81	1.770	.02739	.5649	36.509	+85
86	6	24588.92	170.75	156.05	501.15	1.741	.02743	.5744	36.456	86
87	7	25007.80	173.66	158.96	500.50	1.714	.02747	.5834	36.407	87
88	8	25432.16	176.61	161.91	499.85	1.687	.02751	.5927	36.350	88
89	9	25862.14	179.59	164.89	499.20	1.660	.02754	.6024	36.311	89
+90	550.66	26297.88	182.62	+167.92	498.55	1.634	.02758	.6120	36.258	+90
91	1	26739.88	185.69	170.99	497.89	1.608	.02761	.6219	36.219	91
92	2	27186.56	188.79	174.09	497.24	1.583	.02765	.6317	36.166	92
93	3	27639.43	191.94	177.24	496.59	1.558	.02769	.6418	36.114	93
94	4	28098.26	195.13	180.43	495.94	1.534	.02772	.6518	36.075	94
+95	555.66	28563.00	198.35	+183.65	495.29	1.510	.02776	.6622	36.023	+95
96	6	29033.86	201.62	186.92	494.63	1.486	.02780	.6729	35.971	96
97	7	29510.69	204.94	190.24	493.97	1.463	.02784	.6835	35.919	97
98	8	29993.52	208.29	193.59	493.32	1.442	.02787	.6934	35.881	98
99	9	30482.52	211.68	196.98	492.66	1.419	.02791	.7047	35.829	99
+100	560.66	30977.78	215.12	+200.42	492.01	1.398	.02795	.7153	35.778	+100

PROPERTIES OF SATURATED AMMONIA.

Temperature		Pressure Absolute.		Heat of Vaporization, Thermal Units.	External Heat. Thermal Units.	Internal Heat. Thermal Units.	Volume of Vapor per Lb., Cu. Ft.	Volume of Liquid per Lb., Cu. Ft.	Weight of a Cu. Ft. of Vapor, Pounds.
Degree F.	Absolute.	Lbs. per Sq. Ft.	Lbs. per Sq. In.						
— 40	420.66	1540.9	10.69	579.67	48.23	531.44	24.37	.0234	.0410
— 35	425.66	1773.6	12 31	576.69	48.48	528.21	21.29	.0236	.0467
— 30	430.66	2035.8	14.13	573.69	48.77	524.92	18.66	.0237	.0535
— 25	435.66	2329.5	16.17	570.68	49.06	521.62	16.41	.0238	.0609
— 20	440.66	2657.5	18.45	567.67	49.38	518.29	14.48	.0240	.0690
— 15	445.66	8022.5	20.99	564.64	49.67	514.97	12.81	.0242	.0779
— 10	450.66	3428.0	23.77	561.61	49.99	511.62	11.36	.0243	.0878
— 5	455.66	3877.2	26.93	558.56	50.31	508.25	10.12	.0244	.0988
0	460.66	4373.5	30.37	555.50	50.68	504.82	9.04	.0246	.1109
+ 5	465.66	4920.5	34.17	552.43	50.84	501.59	8.06	.0247	.1241
+ 10	470.66	5522.2	38.55	549.35	51.13	498.22	7.23	.0249	.1384
+ 15	475.66	6182.4	42.93	546.26	51.33	494.93	9.49	.0250	.1540
+ 20	480.66	6905.3	47.95	543.15	51.61	491.54	5.84	.0252	.1712
+ 25	485.66	7695.2	53.43	540.03	51.80	488.23	5.26	.0253	.1901
+ 80	490.66	8556.6	59.41	536.92	52.01	484.91	4.75	.0254	.2105
+ 35	495.66	9493.9	65.93	533.78	52.22	481.56	4.31	.0256	.2320
+ 40	500.66	10512	73.00	530.63	52.42	478.21	3.91	.0257	.2583
+ 45	505.66	11616	80.66	527.47	52.62	474.85	3.56	.0260	.2809
+ 50	510.66	12811	88.96	524.30	52.82	471.48	3.25	.0260	.3109
+ 55	515.66	14102	97.93	521.12	53.01	468.11	2.96	.0260	.3379
+ 60	520.66	15494	107.60	517.93	53.21	464.72	2.70	.0265	.3704
+ 65	525.66	16993	118.03	514.73	53.38	461.35	2.48	.0266	.4034
+ 70	530.66	18605	129.21	511.52	53.57	457.85	2.27	.0268	.4405
+ 75	535.66	20336	141.25	508.29	53.76	454.53	2.08	.0270	.4808
+ 80	540.66	22192	154.11	504.66	53 96	450.70	1.91	.0272	.5262
+ 85	545.66	24178	167.86	501.81	54.15	447.66	1.77	.0273	.5649
+ 90	550.66	26300	182.8	498.11	54.28	443.83	1.64	.0274	.6098
+ 95	555.66	28565	198.37	495.29	54.41	440.88	1.51	.0277	.6622
+100	560.66	30980	215.14	491.50	54.54	436.96	1.39	.0279	.7194
+105	565.66	33550	232.98	488.72	54.67	434.08	1.289	.0281	.7757
+110	570.66	36284	251.97	485.42	54.78	430.64	1.203	.0283	.8312
+115	575.66	39188	272.14	482.41	54.91	427.40	1.121	.0285	.8912
+120	580.66	42267	293.49	478.79	55.03	423.75	1.041	.0287	.9608
+125	585.66	45528	316.16	475.45	55.09	420.39	.9699	.0289	1.0310
+130	590.66	48978	340.42	472.11	55.16	416.94	.9051	.0291	1.1048
+135	595.66	52626	365.16	468.75	55.22	413.53	.8457	.0293	1.1824
+140	600.66	56483	392.22	465.39	55.29	410.09	.7910	.0295	1.2642
+145	605.66	60550	420.49	462.01	55.34	406.67	.7408	.0297	1.3497
+150	610.66	64833	450.20	458.62	55.39	402.23	.6946	.0299	1.4396
+155	615.66	69341	481.54	455.22	55.43	399.79	.6511	.0302	1.5358
+160	620.66	74086	514.40	451.81	55.48	396.35	.6128	.0304	1.6318
+165	625.66	79071	549.04	448.39	55.48	392.94	.5765	.0306	1.7344

The critical pressure of ammonia is 115 atmospheres, the critical temperature at 130° F. (Dewar), critical volume .00482 (calculated).

TABLE OF AMMONIA GAS (SUPER-HEATED VAPOR).

Temperature in Degrees F.

Press. p in Lbs. Abso.	0	5	10	15	20	25	30	35	40	45
No. of Cu. Ft., v, Approximately Contained in 1 Lb. of Gas.										
15	18.81	19.05	19.20	19.48	19.68	19.87	20.06	20.25	20.544	20.74
16	17.56	17.85	18.09	18.24	18.43	18.52	18.81	18.90	19.20	19.44
17	16.60	16.70	16.96	17.08	17.28	17.48	17.66	17.85	18.09	18.31
18	15.54	15.84	15.93	16.12	16.32	16.51	16.70	16.89	17.08	17.32
19	14.78	14.97	15.16	15.26	15.45	15.64	15.84	15.93	16.12	16.36
20	14.01	14.25	14.40	14.49	14.68	14.88	14.97	15.16	15.36	15.58
21	13.34	13.53	13.63	13.82	14.01	14.11	14.30	14.40	14.59	14.80
22	12.76	12.86	13.05	13.15	13 34	13.44	13.63	13.72	13.92	14.12
23	12.19	12.28	12.48	12.57	12.76	12.86	13.05	13.15	13.34	13.54
24	11.71	11.80	11.90	12.09	12.19	12.38	12.48	12.57	12.76	12.96
25	11.23	11.34	11.42	11.61	11.71	11.80	11.90	12 09	12.19	12.38
26	10.75	10.84	11.04	11.13	11.23	11.32	11.52	11.61	11.71	11.85
27	10.36	10.46	10.56	10.75	10.84	10.94	11.04	11.23	11.32	11.45
28	9.98	10.08	10.17	10.36	10.46	10.56	10.65	10.75	10.84	10.94
29	9.60	9.69	9.79	9.98	10.08	10.17	10.27	10.36	10.46	10.57
30	9.2120	9.30	10.46	9.60	9.69	9.79	9.98	10.08	10.17	10.27
31	8.84	9.12	9.21	9.31	9.40	9.50	9.60	9.69	9.80	9.91
32	.....	8.83	8.93	9.02	9.12	9.21	9.31	9.40	9.50	9.61
33	.....	8.54	8.64	8.73	8.83	8.91	9.02	9.11	9.21	9.31
34	.....	8.25	9.35	8.49	8.54	8.64	8.73	8.83	8.92	9.02
35	.....	.....	8.16	8.25	8.35	8.44	8.54	8.64	8.64	8.75
36	.....	.....	7.87	7.96	8.06	8.16	8.26	8.35	8.44	8.55
37	.....	.....	7.68	7.67	7.87	7.96	8.06	8.16	8.26	8.36
38	.....	.....	7.48	7.58	7.68	7.77	7.77	7.87	7.96	8.05
39	.....	.....	.....	7.39	7.48	7.48	7.58	7.68	7.77	7.87
40	.....	.....	.....	7.20	7.29	7.39	7.39	7.48	7.58	7.68
41	.....	.....	.....	7.00	7.10	7.20	7.20	7.29	7.39	7.49
42	.....	.....	.....	6.81	6.91	7.00	7.10	7.10	7.20	7.30
43	.....	.....	.....	.....	6.72	6.81	6.91	7.00	7.08	7.16
44	.....	.....	.....	.....	6.52	6.62	6.72	6 81	6.91	7.10
45	.....	.....	.....	.....	6.43	6.52	6.62	6.62	6.72	6.82

TEMPERATURE AND VOLUME OF STEAM at different pressures, calculated from the experiments of Regnault, etc.

Pressure above the Atmosphere in lbs. per Square Inch.	Temperature Fahr.	Cubic Ft. of Steam from One Cubic Foot of Water.	Pressure above the Atmosphere in lbs. per Square Inch.	Temperature Fahr.	Cubic Ft. of Steam from One Cubic Foot of Water.	Pressure above the Atmosphere in lbs. per Square Inch.	Temperature Fahr.	Cubic Ft. of Steam from One Cubic Foot of Water.
	°			°			°	
0	212	1,640	44	291	446	88	330	264
1	215	1,544	45	292	439	89	330	261
2	219	1,456	46	294	432	90	331	258
3	222	1,378	47	295	426	91	332	256
4	224	1,310	48	296	419	92	333	254
5	227	1,247	49	297	413	93	333	252
6	230	1,190	50	298	407	94	334	250
7	232	1,138	51	299	401	95	334	248
8	235	1,089	52	300	394	96	335	246
9	237	1,048	53	301	387	97	336	244
10	239	1,008	54	302	384	98	336	242
11	241	969	55	303	379	99	337	240
12	244	936	56	304	374	100	338	238
13	246	904	57	305	369	102	339	234
14	248	875	58	305	364	104	340	230
15	250	846	59	306	360	105	341	228
16	252	820	60	307	355	106	342	226
17	253	796	61	308	350	108	343	223
18	255	774	62	309	346	110	344	220
19	257	743	63	310	342	115	347	212
20	259	732	64	311	338	120	350	203
21	260	713	65	312	334	125	353	197
22	262	694	66	313	330	130	355	190
23	264	676	67	314	326	135	358	184
24	265	660	68	314	322	140	361	179
25	267	644	69	315	319	145	363	174
26	268	630	70	316	315	150	366	169
27	270	615	71	317	312	160	370	159
28	271	602	72	318	309	170	375	151
29	272	589	73	318	306	180	380	144
30	274	576	74	319	302	190	384	138
31	275	564	75	320	299	200	388	132
32	277	553	76	321	296	210	392	126
33	278	542	77	322	293	220	396	121
34	279	532	78	322	290	230	399	116
35	281	521	79	323	287	240	403	112
36	282	512	80	324	284	250	406	108
37	283	503	81	325	282	260	409	104
38	284	494	82	325	279	270	413	101
39	285	485	83	326	276	280	416	98
40	286	476	84	327	273	290	419	95
41	288	468	85	328	271	300	422	91
42	289	460	86	328	268	.....	.....	.....
43	290	453	87	329	266	.....	.....	.....

TABLE OF MEAN ABSOLUTE STEAM PRESSURES.

Absolute Initial Pressure in Lbs. per Square Inch	CUT-OFF AND RATIO OF EXPANSION.															
	1/10	1/8	1/6	1/5	1/4	3/10	1/3	4/10	1/2	6/10	2/3	7/10	3/4	8/10	7/8	9/10
	10	8	6	5	4	3.33	3	2.5	2	1.66	1.5	1.43	1.33	1.25	1.14	1.11
10	3.303	3.848	4.648	5.218	5.965	6.609	6.989	7.664	8.465	9.037	9.366	9.489	9.654	9.784	9.921	9.946
15	4.954	5.774	6.980	7.827	8.947	9.913	10.49	11.49	12.69	13.55	14.05	14.23	14.48	14.67	14.88	14.92
20	6.606	7.697	9.297	10.43	11.93	13.21	13.98	15.32	16.93	18.07	18.73	18.98	19.30	19.56	19.84	19.89
25	8.255	9.620	11.63	13.04	14.91	16.52	17.48	19.16	21.16	22.59	23.41	23.72	24.13	24.46	24.80	24.86
30	9.909	11.54	13.96	15.65	17.90	19.82	20.99	22.99	25.39	27.11	28.10	28.46	28.96	29.35	29.76	29.84
35	11.56	13.47	16.28	18.26	20.87	23.13	24.48	26.82	29.64	31.62	32.78	33.21	33.79	34.24	34.72	34.81
40	13.21	15.39	18.59	20.87	23.86	26.43	27.98	30.65	33.86	36.15	37.43	37.95	38.61	39.13	39.68	39.78
45	14.86	17.32	20.94	23.48	26.84	29.74	31.48	34.48	38.09	40.66	42.14	42.70	43.44	44.02	44.64	44.75
50	16.51	19.24	23.26	26.09	29.82	33.04	34.96	38.32	42.32	45.18	46.82	47.44	48.27	48.92	49.60	49.73
60	19.81	23.09	27.92	31.31	35.79	39.65	41.98	45.98	50.79	54.22	56.20	56.93	57.92	58.70	59.52	59.67
70	23.10	26.94	32.55	36.53	41.75	46.26	48.96	53.65	59.25	63.26	65.56	66.42	67.57	68.48	69.44	69.62
80	26.42	30.79	37.21	41.74	47.72	52.87	55.96	61.31	67.72	72.30	74.86	75.91	77.23	78.27	79.37	79.56
90	29.73	34.64	41.88	46.96	53.68	59.48	62.97	68.97	76.18	81.34	84.28	85.40	86.88	88.05	89.28	89.50
100	33.03	38.48	46.51	52.18	59.65	66.08	69.92	76.64	84.65	90.36	93.64	94.89	96.54	97.84	99.21	99.46
110	36.33	42.34	51.17	57.40	65.61	72.70	76.94	84.30	93.11	99.40	103.0	104.4	106.2	107.6	109.1	109.4
120	39.64	46.18	55.84	62.62	71.58	79.31	83.96	91.96	101.6	108.4	112.4	113.8	115.8	117.4	119.0	119.3
130	42.94	50.03	60.47	67.83	77.54	85.91	90.92	99.63	110.0	117.5	121.7	123.3	125.5	127.2	128.9	129.3
140	46.23	53.88	65.13	73.05	83.51	92.52	97.94	107.3	118.5	126.5	131.1	132.8	135.1	136.9	138.8	139.2
150	49.54	57.73	69.80	78.27	89.47	99.13	104.9	114.9	126.9	135.5	140.5	142.3	144.8	146.7	148.8	149.2
200	66.06	76.97	93.05	104.4	119.3	132.2	139.9	153.3	169.3	180.7	187.3	189.8	193.0	195.6	198.4	198.9

NOTE.—To find the mean *effective* pressure, deduct the mean absolute *back* pressure from the pressure given in the table.

TABLE OF PISTON SPEEDS: FEET PER MINUTE.

Stroke in Inches.	REVOLUTIONS PER MINUTE.																								Stroke in Inches.
	50	60	75	80	90	100	120	125	140	150	160	170	180	200	225	240	250	275	300	320	325	350	360	375	400	
6	50	60	75	80	90	100	120	125	140	150	160	170	180	200	225	240	250	275	300	320	325	350	360	375	400	6
7	58	70	87	93	105	117	140	146	163	175	187	198	210	233	262	280	292	321	350	373	379	408	420	437	467	7
8	67	80	100	107	120	133	160	167	187	200	213	227	240	267	300	320	333	367	400	427	433	467	480	500	533	8
9	75	90	112	120	135	150	180	187	210	225	240	255	270	300	337	360	375	412	450	480	487	525	540	562	600	9
10	83	100	125	133	150	167	200	208	233	250	267	283	300	333	375	400	417	458	500	533	542	583	600	625	667	10
11	92	110	137	147	165	183	220	229	257	275	293	312	330	367	412	440	458	504	550	587	596	642	660	687	733	11
12	100	120	150	160	180	200	240	250	280	300	320	340	360	400	450	480	500	550	600	640	650	700	720	750	800	12
14	117	140	175	187	210	233	280	292	303	350	373	397	420	466	525	560	583	642	700	747	758	817	840	875	933	14
15	125	150	187	200	225	250	300	312	350	375	400	425	450	500	562	600	625	687	750	800	812	875	900	937	1000	15
16	133	160	200	213	240	267	320	333	373	400	427	453	480	533	600	640	667	733	800	853	867	933	960	1000	1067	16
18	150	180	225	240	270	300	360	375	420	450	480	510	540	600	675	720	750	825	900	960	975	1050	1080	1125	1200	18
20	167	200	250	267	300	334	400	417	467	500	533	587	600	667	750	800	833	917	1000	1067	1083	1167	1200	1250	1333	20
22	193	220	275	293	330	367	440	458	513	550	587	623	660	733	825	880	917	1008	1100	1173	1192	1233	1320	1375	1467	22
24	200	240	300	320	360	400	480	500	560	600	640	680	720	800	900	960	1000	1100	1200	1280	1300	1400	1440	1500	1600	24
26	217	260	325	347	390	433	520	542	607	650	693	733	780	867	975	1040	1083	1192	1300	1360	1408	1517	1560	1625	1731	26
28	233	280	350	373	420	467	560	583	653	700	746	793	840	933	1050	1120	1166	1283	1400	1494	1516	1634	1680	1750	1866	28
30	250	300	375	400	450	500	600	625	700	750	800	850	900	1000	1125	1200	1250	1375	1500	1600	1624	1750	1800	1874	2000	30
36	300	360	450	480	540	600	720	750	840	900	960	1020	1080	1200	1350	1440	1500	1650	1800	1920	1950	2100	2160	2250	2400	36
40	333	400	500	533	600	667	800	833	933	1000	1066	1133	1200	1333	1500	1600	1666	1834	2000	2134	2166	2334	2400	2500	2666	40
42	350	420	525	560	630	700	840	875	980	1050	1120	1190	1360	1400	1575	1680	1749	1926	2100	2241	2274	2451	2520	2625	2799	42
48	400	480	600	640	720	800	960	1000	1120	1200	1300	1260	1440	1600	1800	1920	2000	2200	2400	2560	2600	2800	2880	3000	3200	48
54	450	540	675	720	810	900	1080	1125	1260	1350	1440	1530	1620	1800	2025	2160	2250	2475	2700	2880	2925	3150	3240	3375	3600	54
60	500	600	750	800	900	1000	1200	1250	1400	1500	1600	1700	1800	2000	2250	2400	2500	2750	3000	3200	3248	3500	3600	3748	4000	60
66	550	660	825	880	990	1100	1320	1375	1540	1650	1761	1869	1980	2199	2475	2640	2751	3024	3300	3519	3576	3699	3960	4125	4401	66
72	600	720	900	960	1080	1200	1440	1500	1680	1800	1920	2040	2160	2400	2700	2880	3000	3300	3600	3840	3900	4200	4320	4500	4800	72

WROUGHT IRON WELDED STEAM, GAS AND WATER PIPE.

TABLE OF STANDARD DIMENSIONS.

Diameter.			Thick-ness.	Circumference.		Transverse Areas.			Length of Pipe per Square Foot of		Length of Pipe cont'ing one Cub. Foot.	Nomin'l weight per Foot.	Number of Threads pr. In. of Screw
Nominal Internal.	Actual External	Actual Internal.		External	Internal.	External.	Internal.	Metal.	External Surface.	Internal Surface.			
Inches.	Inches.	Inches.	Ins.	Inches.	Inches.	Sq. Ins.	Sq. Ins.	Sq. Ins.	Feet.	Feet.	Feet	Pounds	
⅛	.405	.27	.068	1.272	.848	.129	.0573	.0717	9.44	14.15	2513.	.241	27
¼	.54	.364	.088	1.696	1.144	.229	.1041	.1249	7.075	10.49	1383.3	.42	18
⅜	.675	.494	.091	2.121	1.552	.358	.1917	.1663	5.657	7.73	751.2	.559	18
½	.84	.623	.109	2.639	1.957	.554	.3048	.2492	4.547	6.13	472.4	.837	14
¾	1.05	.824	.113	3.299	2.589	.866	.5333	.3327	3.637	4.635	270.	1.115	14
1	1.315	1.048	.134	4.131	3.292	1 358	.8626	.4954	2.904	3.645	166.9	1.668	11½
1¼	1.66	1.380	.14	5.215	4.335	2.164	1.496	.668	2.301	2.768	96.25	2.244	11½
1½	1.9	1.611	.145	5.969	5.061	2.835	2.038	.797	2.01	2.371	70.66	2.678	11½
2	2.375	2.067	.154	7.461	6.494	4.43	3.356	1.074	1.608	1.848	42.91	3.609	11½
2½	2.875	2.468	.204	9.032	7.753	6.492	4.784	1.708	1.328	1.547	30.1	5.739	8
3	3.5	3.067	.217	10.996	9.636	9.621	7.388	2.243	1.091	1.245	19.5	7.536	8
3½	4.	3.548	.226	12.566	11.146	12.566	9.887	2.679	.955	1.077	14.57	9.001	8
4	4.5	4.026	.237	14.137	12.648	15.904	12.73	3.174	.849	.949	11.31	10.665	8
4½	5.	4.508	.246	15.708	14.162	19.635	15.961	3.674	.764	.848	9.02	12.49	8
5	5.563	5.045	.259	17.477	15.849	24.306	19.99	4.316	.687	.757	7.2	14.502	8
6	6.625	6.065	.28	20.813	19.054	34.472	28.888	5.584	.577	.63	4.08	18.762	8
7	7.625	7.023	.301	23.955	22.063	45.664	38.738	6.926	.501	.544	3.72	23.271	8
8	8.625	7.982	.322	27.096	25.076	58.426	50.04	8.386	.443	.478	2.88	28.177	8
9	9.625	8.937	.344	30.238	28.076	72.76	62.73	10.03	.397	.427	2.29	33.701	8
10	10.75	10.019	.366	33.772	31.477	90.763	78.839	11.924	.355	.382	1.82	40.065	8
11	11.75	11.	.....	36.914	34.558	108.434	95.033	13.401	.325	.347	1.51	45.028	8
12	12.75	12.	....	40.055	37.7	127.677	113.098	14.579	.299	.319	1.27	48.985	8
13	14.	13.25	.....	43.982	41.626	153.938	137.887	16.051	.273	.288	1.04	53.921	8
14	15.	14.25	.....	47.124	44.768	176.715	159.485	17.23	.255	.268	.903	57.893	8
15	16.	15.43	.....	50.26	48.48	201.06	187.04	14.02	.239	.248	.77	62.	8
16	17.	16.4	.....	53.41	51.52	226.98	211.24	15.74	.225	.238	.68	.....	8
17	18.	17.32	.....	56.55	54.41	254.47	235.61	18.86	.212	.221	.61	.....	8

STANDARD DIMENSIONS WROUGHT IRON WELDED EXTRA STRONG PIPE.

Diameter.			Thickness.	Nearest Wire Gauge.	Circumference.		Transverse Areas.			Length of Pipe per Square Foot of		Nominal Weight per Foot
Nominal Internal.	Actual External	Actual Internal.			External	Internal.	External.	Internal.	Metal.	External Surface.	Internal Surface.	
Inches.	Inches.	Inches.	Ins.	No.	Inches.	Inches.	Sq. Inches	Sq. Inches	Sq. Inches	Feet.	Feet.	Pounds
⅛	.405	.205	.1	12½	1.272	.644	.129	.033	.086	9.433	18.632	.29
¼	.54	.294	.123	11	1.696	.924	.229	.068	.161	7.075	12.986	.54
⅜	.675	.421	.127	10½	2.121	1.323	.358	.139	.219	5.657	9.07	.74
½	.84	.542	.149	9	2.639	1.703	.554	.231	.323	4.547	7.046	1.09
¾	1.05	.736	.157	8½	3.299	2.312	.866	.452	.414	3.637	5.109	1.39
1	1.315	.951	.182	7	4.131	2.988	1.358	.71	.648	2.904	4.016	2.17
1¼	1.66	1.272	.194	6½	5.215	3.996	2.164	1.271	.893	2.301	3.003	3.
1½	1.9	1.494	.203	6	5.969	4.694	2.835	1.753	1.082	2.01	2.556	3.63
2	2.375	1.933	.221	5	7.461	6.073	4.43	2.935	1.495	1.608	1.975	5.02
2½	2.875	2.315	.28	2	9.032	7.273	6.492	4.209	2.283	1.328	1.649	7.67
3	3.5	2.892	.304	1	10.996	9.085	9.621	6.569	3.052	1.091	1.328	10.25
3½	4.	3.358	.321	0	12.566	10.549	12.566	8.856	3.71	.955	1.137	12.47
4	4.5	3.818	.341	0	14.137	11.995	15.904	11.449	4.455	.849	1.	14.97
5	5.563	4.813	.375	00	17.477	15.120	24.306	18.193	6.12	.687	.793	20.54
6	6.625	5.75	.437	000	20.813	18.064	34.472	25.967	8.505	.577	.664	28.58

DOUBLE EXTRA STRONG PIPE.

½	.84	.244	.298	1	2.639	.766	.554	.047	.507	4.547	15.667	1.7
¾	1.05	.422	.314	1	3.299	1.326	.866	.139	.727	3.637	9.049	2.44
1	1.315	.587	.364	00	4.131	1.844	1.358	.271	1.087	2.904	6.508	3.65
1¼	1.66	.885	.388	00	5.215	2.78	2.164	.615	1 549	2.304	4.317	5.2
1½	1.9	1.068	.406	000	5.969	3.418	2.835	.93	1.905	2.01	3 511	6.4
2	2.375	1.491	.442	0000	7.461	4.684	4.43	1.744	2.686	1.608	2.561	9.02
2½	2.875	1.755	.560	1/8—	9.032	5.513	6.492	2.419	4.073	1.328	2.176	13.68
3	3.5	2.284	.608	3/8—	10.996	7.175	9.621	4.097	5.524	1.091	1.672	18.56
3½	4.	2.716	.642	5/8+	12.566	8.533	12.566	5.794	6.772	.955	1.406	22.75
4	4.5	3.136	.682	1/8—	14.137	9.852	15.904	7.724	8.18	.849	1.217	27.48
5	5.563	4.063	.75	¾	17.477	12.764	24.306	12.965	11.34	.687	.940	38 12

HEAD OF WATER AND EQUIVALENT PRESSURE IN POUNDS PER SQUARE INCH.

Head in ft.	Press.	Head in ft.	Press.	Head in ft.	Press.	Head in ft.	Press.	Head in ft.	Press.
1	0.43	41	17.75	81	35.08	121	52.41	161	69.74
2	0.86	42	18.19	82	35.52	122	52.84	162	70.17
3	1.30	43	18.62	83	35.95	123	53.28	163	70.61
4	1.73	44	19.05	84	36.39	124	53.71	164	71.04
5	2.16	45	19.49	85	36.82	125	54.15	165	71.47
6	2.59	46	19.92	86	37.25	126	54.58	166	71.91
7	3.03	47	20.35	87	37.68	127	55.01	167	72.34
8	3.46	48	20.79	88	38.12	128	55.44	168	72.77
9	3.89	49	21.22	89	38.55	129	55.88	169	73.20
10	4.33	50	21.65	90	39.98	130	56.31	170	73.64
11	4.76	51	22.09	91	39.42	131	56.74	171	74.07
12	5.20	52	22.52	92	39.85	132	57.18	172	74.50
13	5.63	53	22.95	93	40.28	133	57.61	173	74.94
14	6.06	54	23.39	94	40.72	134	58.04	174	75.37
15	6.49	55	23.82	95	41.15	135	58.48	175	75.80
16	6.93	56	24.26	96	41.58	136	58.91	176	76.23
17	7.36	57	24.69	97	42.01	137	59.34	177	76.67
18	7.79	58	25.12	98	42.45	138	59.77	178	77.10
19	8.22	59	25.55	99	42.88	139	60.21	179	77.53
20	8.66	60	25.99	100	43.31	140	60.64	180	77.97
21	9.09	61	26.42	101	43.75	141	61.07	181	78.40
22	9.53	62	26.85	102	44.18	142	61.51	182	78.84
23	9.96	63	27.29	103	44.61	143	61.94	183	79.27
24	10.39	64	27.72	104	45.05	144	62.37	184	79.70
25	10.82	65	28.15	105	45.48	145	62.81	185	80.14
26	11.26	66	28.58	106	45.91	146	63.24	186	80.57
27	11.69	67	29.02	107	46.34	147	63.67	187	81.00
28	12.12	68	29.45	108	46.78	148	64.10	188	81.43
29	12.55	69	29.88	109	47.21	149	64.54	189	81.87
30	12.99	70	30.32	110	47.64	150	64.97	190	82.30
31	13.42	71	30.75	111	48.08	151	65.49	191	82.73
32	13.86	72	31.18	112	48.51	152	65.84	192	83.17
33	14.29	73	31.62	113	48.94	153	66.27	193	83.60
34	14.72	74	32.05	114	49.38	154	66.70	194	84.03
35	15.16	75	32.48	115	49.81	155	67.14	195	84.47
36	15.59	76	32.92	116	50.24	156	67.57	196	84.90
37	16.02	77	33.35	117	50.68	157	68.00	197	85.33
38	16.45	78	33.78	118	51.11	158	68.43	198	85.76
39	16.89	79	34.21	119	51.54	159	68.87	199	86.20
40	17.32	89	34.65	120	51.98	160	69.31	200	86.63

CAPACITIES OF DOUBLE-SUCTION CENTRIFUGAL PUMPS.

(HEALD & SISCO PATENTS.)

Size Discharge Opening, Inches.	Size of Pipe Recommended.		Capacity in Gallons per Minute.	H. P. Required for Each Ft. of Lift, Min. Quantity.	Diameter and Face of Pulley, in Inches.	Floor Space Required, in Inches, about.	Shipping Weight, Pounds.
	Suction.	Discharge.					
No. 1½	2	2	50 to 70	.024	7x 8	20x32	250
1¾	2½	2	75 to 100	.037	7x 8	20x32	300
2	3	2½	110 to 150	.054	8x 8	26x35	500
2½	3½	3	175 to 250	.086	8x 8	26x35	550
3	4	3½	250 to 350	.124	8x 8	27x38	575
4	5	4½	450 to 600	.223	10x10	33x40	800
5	6	5	750 to 900	.372	15x12	37x49	1,300
6	8	6	1,000 to 1,400	.496	15x12	43x51	1,450
8	10	8	1,700 to 2,200	.844	20x12	52x61	2,190
10	12	10	2,200 to 3,000	1.093	24x12	57x73	3,000
12	15	12	3,000 to 4,000	1.49	30x14	69x82	5,000
15	18	15	4,800 to 6,000	2.38	40x15	89x78	8,000
18	20	18	7,500 to 10,000	3.73	40x15	90x80	9,000

CAPACITIES OF STANDARD HORIZONTAL CENTRIFUGAL PUMPS.

No. of Pump.	Capacity in Gallons per Minute.	H.P. Required for Each Foot of Lift, Min. Quantity.	Diameter and Face of Pulley, in Inches	Floor Space Required, in Inches.	Shipping Weight, Pounds.	No. of Pump.
1½	50 to 70	.024	6x 6	17x 30	168	1½
1¾	75 to 100	.037	7x 8	21x 33	232	1¾
2	110 to 150	.054	8x 8	23x 37	306	2
2½	175 to 250	.086	8x 8	24x 38	348	2½
3	250 to 350	.124	8x 8	25x 39	400	3
4	450 to 600	.223	10x10	30x 40	545	4
5	750 to 900	.372	15x12	34x 54	826	5
6	1,000 to 1,400	.496	15x12	37x 55	965	6
8	1,700 to 2,200	.844	20x12	45x 63	1.500	8
10	2,200 to 3,000	1.093	24x12	51x 71	2,170	10
12	3,000 to 4,000	1.49	30x14	62x 78	3,050	12
15	4,800 to 6,000	2.38	40x15	77x 80	7,100	15
*15	4,800 to 6,000	2.38	30x15	60x 68	3,150	*15
18	7,500 to 10,000	3.73	40x15	93x103	9,000	18
*18	7.500 to 10,000	3.73	30x16	62x 70	3.500	*18
22	12.000 to 14,000	5.96	48x20	126x130	12,000	22

* Refers to low-lift pump.

The number of pump is also diameter of discharge opening in inches.

COMPARISONS OF THERMOMETER SCALES, SHOWING THE RELATIVE INDICATIONS OF THE CELSIUS, FAHRENHEIT AND REAUMUR THERMOMETER SCALES.

In the United States and England the Fahrenheit scale is generally used; in France and in all scientific investigations and treatises, the Celsius scale is uniformly used; and in Germany the Reaumur scale is the one generally adopted.

C.	F.	R.	C.	F.	R.	C.	F.	R.
100°	212.0°	80.0°	53°	127.4°	42.4°	6°	42.8°	4.8°
99	210.2	79.2	52	125.6	41.6	5	41.0	4.0
98	208.4	78.4	51	123.8	40 8	4	39.2	3.2
97	206.6	77.6	50	122.0	40.0	3	37.4	2.4
96	204.8	76.8	49	120.2	39.2	2	35.6	1.6
95	203.0	76.0	48	118.4	38.4	1	33.8	0.8
94	201.2	75.2	47	116.6	37.6	Zero	32.0	Zero
93	199.4	74.4	46	114.8	36.8	1	30.2	0.8
92	197.6	73.6	45	113.0	36.0	2	28.4	1.6
91	195.8	72.8	44	111.2	35.2	3	26.6	2.4
90	194.0	72.0	43	109.4	34.4	4	24.8	3.2
89	192.2	71.2	42	107.6	33.6	5	23.0	4.0
88	190.4	70.4	41	105.8	32.8	6	21.2	4.8
87	188.6	69.6	40	104.0	32.0	7	19.4	5.6
86	186.8	68.8	39	102.2	31.2	8	17.6	6.4
85	185.0	68.0	38	100.4	30.4	9	15.8	7.2
84	183.2	67.2	37	98.6	29.6	10	14.0	8.0
83	181.4	66.4	36	96.8	28.8	11	12.2	8.8
82	179.6	65.6	35	95.0	28.0	12	10.4	9.6
81	177.8	64.8	34	93.2	27.2	13	8.6	10.4
80	176.0	64.0	33	91.4	26.4	14	6.8	11.2
79	174.2	63.2	32	89.6	25.6	15	5.0	12.0
78	172.4	62.4	31	87.8	24.8	16	3.2	12.8
77	170.6	61.6	30	86.0	24.0	17	1.4	13.6
76	168.8	60.8	29	84.2	23.2	18	—	14.4
75	167.0	60.0	28	82.4	22.4	19	2.2	15.2
74	165.2	59.2	27	80.6	21.6	20	4.0	16.0
73	163.4	58.4	26	78.8	20.8	21	5.8	16.8
72	161.6	57.6	25	77.0	20.0	22	7.6	17.6
71	159.8	56.8	24	75.2	19.2	23	9.4	18.4
70	158.0	56.0	23	73.4	18.4	24	11.2	19.2
69	156.2	55.2	22	71.6	17.6	25	13.0	20.0
68	154.4	54.4	21	69.8	16.8	26	14.8	20.8
67	152.6	53.6	20	68.0	16.0	27	16.6	21.6
66	150.8	52.8	19	66.2	15.2	28	18.4	22.4
65	149.0	52.0	18	64.4	14.4	29	20.2	23.2
64	147.2	51.2	17	62.6	13.6	30	22.0	24.0
63	145.4	50.4	16	60.8	12.8	31	23.8	24.8
62	143.6	49.6	15	59.0	12.0	32	25.6	25.6
61	141.8	48.8	14	57.2	11.2	33	27.4	26.4
60	140.0	48.0	13	55.4	10.4	34	29.2	27.2
59	138.2	47.2	12	53.6	9.6	35	31.0	28.0
58	136.4	46.4	11	51.8	8.8	36	32.8	28.8
57	134.3	45.6	10	50.0	8.0	37	34.6	29.6
56	132.8	44.8	9	48.2	7.2	38	36.4	30.4
55	131.0	44.0	8	46.4	6.4	39	38.2	31.2
54	129.2	43.2	7	44.6	5.8	40	40.0	32.0

DIAMETERS, AREAS AND CIRCUMFERENCES OF CIRCLES.

Diam. Inches.	Circumf. Inches.	Area. Sq. In.	Diam. Inches.	Circumf. Inches.	Area. Sq. In.	Diam. Inches.	Circumf. Inches.	Area. Sq. In.
1	3.14159	0.7854	4	12.5664	12.566	8	25.1327	50.265
1/16	3.33794	0.8866	1/16	12.7627	12.962	1/8	25 5224	51.849
1/8	3.53429	0.9940	1/8	12.9591	13.364	1/4	25.9181	53.456
3/16	3.73064	1.1075	3/16	13.1554	13.772	3/8	26.3108	55.088
1/4	3.92699	1.2272	1/4	13.3518	14.186	1/2	26.7035	56.745
5/16	4.12334	1.3530	5/16	13.5481	14.607	5/8	27.0962	58.426
3/8	4.31969	1.4849	3/8	13.7445	15.033	3/4	27.4889	60.132
7/16	4.51604	1.6230	7/16	13.9408	15.466	7/8	27.8816	61.862
1/2	4.71239	1.7671	1/2	14.1372	15.904	9	28.2743	63.617
9/16	4.90874	1.9175	9/16	14.3335	16.349	1/8	28.6670	65.397
5/8	5.10509	2.0739	5/8	14.5299	16.800	1/4	29.0597	67.201
11/16	5.30144	2.2365	11/16	14.7262	17.257	3/8	29.4524	69.029
3/4	5.49779	2.4053	3/4	14.9226	17.721	1/2	29.8451	70.882
13/16	5.69414	2.5802	13/16	15.1189	18.190	5/8	30.2378	72.760
7/8	5.89049	2.7612	7/8	15.3153	18.665	3/4	30.6305	74.662
15/16	6.08684	2.9483	15/16	15.5116	19.147	7/8	31.0232	76.589
2	6.28319	3.1416	5	15.7080	19.635	10	31.4159	78.540
1/16	6.47953	3.3410	1/16	15.9043	20.129	1/4	32.2013	82.516
1/8	6.67588	3.5466	1/8	16.1007	20.629	1/2	32.9867	86.590
3/16	6.87223	3.7583	3/16	16.2970	21.135	3/4	33.7721	90.763
1/4	7.06858	3.9761	1/4	16.4934	21.648	11	34.5575	95.033
5/16	7.26493	4.2000	5/16	16.6897	22.166	1/4	35.3429	99.402
3/8	7.46128	4.4301	3/8	16.8861	22.691	1/2	36.1283	103.87
7/16	7.65763	4.6664	7/16	17.0824	23.221	3/4	36.9137	108.43
1/2	7.85398	4.9087	1/2	17.2788	23.758	12	37.6991	113.10
9/16	8.05033	5.1572	9/16	17.4751	24.301	1/4	38.4845	117.86
5/8	8.24668	5.4119	5/8	17.6715	24.850	1/2	39.2699	122.72
11/16	8.44303	5.6727	11/16	17.8678	25.406	3/4	40.0553	127.68
3/4	8.63938	5.9396	3/4	18.0642	25.967	13	40.8407	132.73
13/16	8.83573	6.2126	13/16	18.2605	26.535	1/4	41.6261	137.89
7/8	9.03208	6.4918	7/8	18.4569	27.109	1/2	42.4115	143.14
15/16	9.22843	6.7771	15/16	18.6532	27.688	3/4	43.1969	148.49
3	9.42478	7.0686	6	18.8496	28.274	14	43.9823	153.94
1/16	9.62113	7.3662	1/16	19.2423	29.465	1/4	44.7677	159.48
1/8	9.81748	7.6699	1/4	19.6350	30.680	1/2	45.5531	165.13
3/16	10.0138	7.9798	3/8	20.0277	31.919	3/4	46.3385	170.87
1/4	10.2102	8.2958	1/2	20.4204	33.183	15	47.1239	176.71
5/16	10.4065	8.6179	5/8	20.8131	34.472	1/4	47.9093	182.65
3/8	10.6029	8.9462	3/4	21.2058	35.785	1/2	48.6947	188.69
7/16	10.7992	9.2806	7/8	21.5984	37.122	3/4	49.4801	194.83
1/2	10.9956	9.6211	7	21.9911	38.485	16	50.2655	201.06
9/16	11.1919	9.9678	1/8	22.3838	39.871	1/4	51.0509	207.39
5/8	11.3883	10.321	1/4	22.7765	41.282	1/2	51.8363	213.82
11/16	11.5846	10.680	3/8	23.1692	42.718	3/4	52.6217	220.35
3/4	11.7810	11.045	1/2	23.5619	44.179	17	53.4071	226.98
13/16	11.9772	11.416	5/8	23.9546	45.664	1/4	54.1925	233.71
7/8	12.1737	11.793	3/4	24.3473	47.173	1/2	54.9779	240.53
15/16	12.3700	12.177	7/8	24.7400	48.707	3/4	65.7633	247.45

Diameters, Areas and Circumferences of Circles.—Con.

Diam. Inches.	Circumf. Inches.	Area. Sq. In.	Diam. Inches.	Circumf. Inches.	Area. Sq. In.	Diam. Inches.	Circumf. Inches.	Area. Sq. In.
18	56.5487	254.47	32	100.531	804.25	46	144.513	1661.9
¼	57.3341	261.59	¼	101.316	816.86	¼	145.299	1680.0
½	58.1195	268 80	½	102.102	829.58	½	146.084	1698.2
¾	58.9049	276.12	¾	102.887	842.39	¾	146.869	1716.5
19	59.6903	283.53	33	103.673	855.30	47	147.655	1734.9
¼	60.4757	291.04	¼	104.458	868.31	¼	148.440	1753.5
½	61.2611	298.65	½	105.243	881.41	½	149.226	1772.1
¾	62.0465	306.35	¾	106.029	894.62	¾	150.011	1790.8
20	62.8319	314.16	34	106.814	907.92	48	150.796	1809 6
¼	63.6173	322.06	¼	107.600	921.32	¼	151.582	1828.5
½	64.4026	330.06	½	108.385	934.82	½	152.367	1847.5
¾	66.1880	338.16	¾	109.170	948.42	¾	153.153	1866.5
21	65.9734	346.36	35	109.956	962.11	49	153.938	1885.7
¼	66.7588	354.66	¼	110.741	975.91	¼	154.723	1905.0
½	67.5442	363.05	½	111.527	989.80	½	155.509	1924.2
¾	68.3296	371.54	¾	112.312	1003.8	¾	156.294	1943.9
22	69.1150	380.13	36	113.007	1017.9	50	157.080	1963.5
¼	69.9004	388.82	¼	113.883	1032.1	¼	157.865	1983.2
½	70.6858	397.61	½	114.668	1046.3	½	158.650	2003.0
¾	71 4712	406.49	¾	115.454	1060.7	¾	159.436	2022.8
23	72.2566	415.48	37	116.239	1075.2	51	160.221	2042.8
¼	73.0420	424.56	¼	117.024	1089.8	¼	161.007	2062.9
½	73.8274	433.74	½	117.810	1104.5	½	161.792	2083.1
¾	74.6128	443.01	¾	118.596	1119.2	¾	162 577	2103.3
24	75.3982	452.39	38	119.381	1134.1	52	163.363	2123.7
¼	76.1836	461.86	¼	120.166	1149.1	¼	164.148	2144.2
½	76.9690	471.44	½	120.951	1164.2	½	164.934	2164.8
¾	77.7544	481.11	¾	121.737	1179.3	¾	165.719	2185.4
25	78.5398	490.87	39	122.522	1194.6	53	166.504	2206.2
¼	79.3252	500.74	¼	123.308	1210.0	¼	167.290	2227.0
½	80.1106	510.71	½	124.093	1225.4	½	168.075	2248.0
¾	80.8960	520.77	¾	124.878	1241.0	¾	168.861	2269.1
26	81.6814	530.93	40	125.664	1256.6	54	169.646	2290.2
¼	82.4668	541.19	¼	126.449	1272.4	¼	170.431	2311.5
½	83.2522	551.55	½	127.235	1288.2	½	171.217	2332.8
¾	84.0376	562.00	¾	128.020	1304.2	¾	172.002	2354.3
27	84.8230	572.56	41	128.805	1320.3	55	172.788	2375.8
¼	85.6084	583.21	¼	129.591	1336.4	¼	173.573	2397.5
½	86.3938	593.96	½	130.376	1352.7	½	174.358	2419.2
¾	87.1792	604.81	¾	131.161	1369.0	¾	175.144	2441.1
28	87.9646	615.75	42	131.947	1385.4	56	175.929	2463.0
¼	88.7500	626.80	¼	132.732	1402.0	¼	176.715	2485.0
½	89.5354	637.94	½	133.518	1418.6	½	177.500	2507.2
¾	90.3208	649.18	¾	134.303	1435.4	¾	178.285	2529.4
29	91.1062	660.52	43	135.088	1452.2	57	179.071	2551.8
¼	91.8916	671.96	¼	135.874	1469.1	¼	179.856	2574.2
½	92.6770	683.49	½	136.659	1486.2	½	180.642	2596.7
¾	93.4624	695.13	¾	137.445	1503.3	¾	181.427	2619.4
30	94.2478	706.86	44	138.230	1520.5	58	182.212	2642.1
¼	95.0332	718.69	¼	139.015	1537.9	¼	182.998	2664.9
½	95.8186	730.62	½	139.801	1555.3	½	183.783	2687.8
¾	96.6040	742.64	¾	140.586	1572.8	¾	184.569	2710.9
31	97.3894	754.77	45	141.372	1590.4	59	185.354	2734.0
¼	98.1748	766.99	¼	142.157	1608.2	¼	186.139	2757.2
½	98.9602	779.31	½	142.942	1626.0	½	186.925	2780.5
¾	99.7456	791.73	¾	143.728	1643.9	¾	187.710	2803.9

Diameters, Areas and Circumferences of Circles.—Con.

Diam. Inches.	Circumf. Inches.	Area. Sq. In.	Diam. Inches.	Circumf. Inches.	Area. Sq. In.	Diam. Inches.	Circumf. Inches.	Area. Sq. In.
60	188.496	2827.4	74	232.478	4300.8	88	276.460	6082.1
¼	189.281	2851.0	¼	233.263	4329.9	¼	277.246	6116.7
½	190.066	2874.8	½	234.049	4359.2	½	278.031	6151.4
¾	190.852	2898.6	¾	234.834	4388.5	¾	278.816	6186.2
61	191.637	2922.5	75	235.619	4417.9	89	279.602	6221.1
¼	192.423	2946.5	¼	236.405	4447.4	¼	280.387	6256.1
½	193.208	2970.6	½	237.190	4477.0	½	281.173	6291.2
¾	193.993	2994.8	¾	237.976	4506.7	¾	281.958	6326.4
62	194.779	3019.1	76	238.761	4536.5	90	282.743	6361.7
¼	195.564	3043.5	¼	239.546	4566.4	¼	283.529	6397.1
½	196.350	3068.0	½	240.332	4596.3	½	284.314	6432.6
¾	197.135	3092.6	¾	241.117	4626.4	¾	285.100	6468.2
63	197.920	3117.2	77	241.903	4656.6	91	285.885	6503.9
¼	198.706	3142.0	¼	242.688	4686.9	¼	286.670	6539.7
½	199.491	3166.9	½	243.473	4717.3	½	287.456	6575 5
¾	200.277	3191.9	¾	244.259	4747.8	¾	288.241	6611.5
64	201.062	3217.0	78	245.044	4778.4	92	289.027	6647.6
¼	201.847	3242.2	¼	245.830	4809.0	¼	289.812	6683.8
½	202.633	3267.5	½	246.615	4839.8	½	290.597	6720.1
¾	203.418	3292.8	¾	247.400	4870.7	¾	291.383	6756.4
65	204.204	3318.3	79	248.186	4901.7	93	292.168	6792.9
¼	204.989	3343.9	¼	248.971	4932.7	¼	292.954	6829.5
½	205.774	3369.6	½	249.757	4963.9	½	293.739	6866.1
¾	206.560	3395.3	¾	250.542	4995.2	¾	294.524	6902.9
66	207.345	3421.2	80	251.327	5026.5	94	295.310	6939.8
¼	208.131	3447.2	¼	252.113	5058.0	¼	296.095	6976.7
½	208.916	3473.2	½	252.898	5089.6	½	296.881	7013.8
¾	209.701	3499.4	¾	253.684	5121.2	¾	297.666	7051.0
67	210.487	3525.7	81	254.469	5153.0	95	298.451	7088.2
¼	211.272	3552.0	¼	255.254	5184.9	¼	299.237	7125.6
½	212.058	3578.5	½	256.040	5216.8	½	300.022	7163.0
¾	212.843	3605.0	¾	256.825	5248.9	¾	300.807	7200.6
68	213.628	3631.7	82	257.611	5281.0	96	301.593	7238.2
¼	214.414	3658.4	¼	258.396	5313.3	¼	302.378	7276.0
½	215.199	3685.3	½	259.181	5345.6	½	303.164	7313.8
¾	215.984	3712.2	¾	259.967	5378.1	¾	303.949	7351.8
69	216.770	3739.3	83	260.752	5410.6	97	304.734	7389.8
¼	217.555	3766.4	¼	261.538	5443.3	¼	305.520	7428.0
½	218.341	3793.7	½	262.323	5476.0	½	306.305	7466.2
¾	219.126	3821.0	¾	263.108	5508.8	¾	307.091	7504.5
70	219.911	3848.5	84	263.894	5541.8	98	307.876	7543.0
¼	220.697	3876.0	¼	264.679	5574.8	¼	308.661	7581.5
½	221.482	3903.6	½	265.465	5607.9	½	309.447	7620.1
¾	222.268	3931.4	¾	266.250	5641.2	¾	310.232	7658.9
71	223.053	3959.2	85	267.035	5674.5	99	311.018	7697.7
¼	223.838	3987.1	¼	267.821	5707.9	¼	311.803	7736.6
½	224.624	4015.2	½	268.606	5741.5	½	312.588	7775.6
¾	225.409	4043.3	¾	269.392	5775.1	¾	313.374	7814.8
72	226.195	4071.5	86	270.177	5808.8	100	314.159	7854.0
¼	226.980	4099.8	¼	270.962	5842.6			
½	227.765	4128.2	½	271.748	5876.5			
¾	228.551	4156.8	¾	272.533	5910.6			
73	229.336	4185.4	87	273.319	5944.7			
¼	230.122	4214.1	¼	274.104	5978.9			
½	230.907	4242.9	½	274.889	6013.2			
¾	231.692	4271.8	¾	275.675	6047.6			

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